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Heating · Piping and Air Conditioning

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INDUSTRY AND
RGE BUILDINGS

April, 1935

IN THIS ISSUE

Design of Modern Industrial Piping Systems--Flexibility

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Apartment Building Air Conditioning

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Installing Piping in Cramped Quarters

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One of 1934's Biggest Jobs

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Choosing the Right Air Conditioning System

and others. Also the

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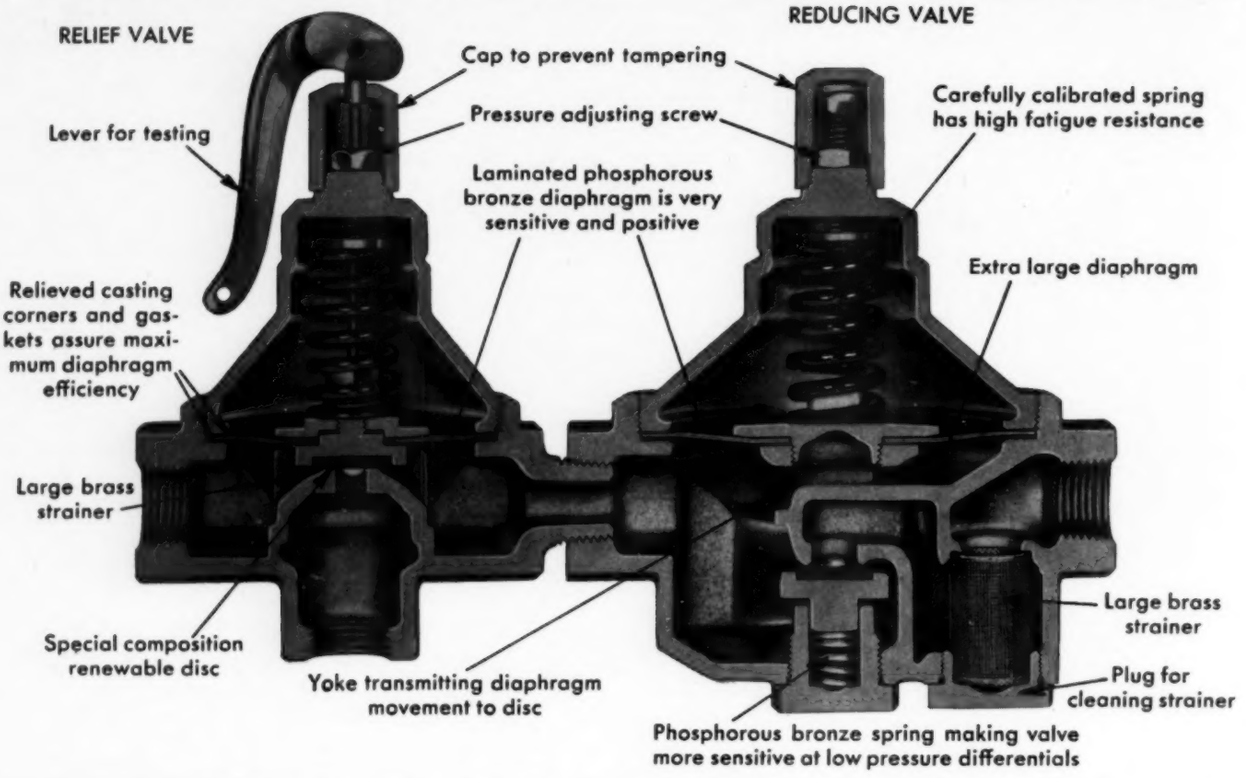
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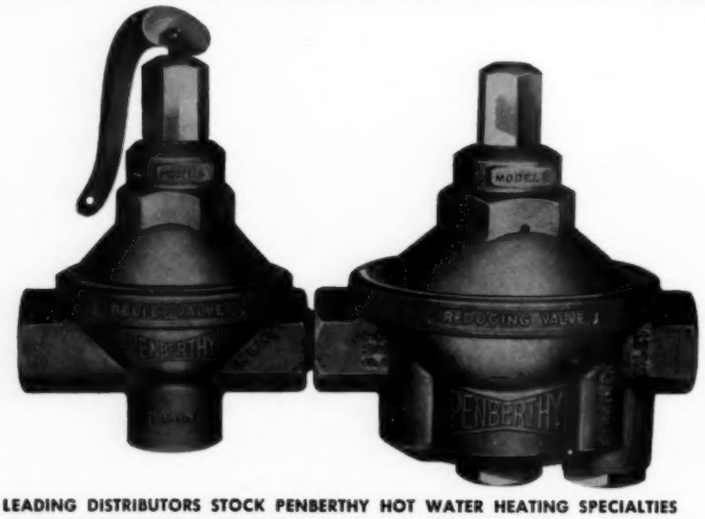

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THE EDITOR'S PAGE

ATTENTION to **plant and building heating** systems now to improve operation next winter is discussed by T. Napier Adlam this month in an article containing many practical suggestions . . . F. L. Snyder writes in this issue on **flexibility of piping;** his article on flow of fluids in January created a lot of interest, many readers requesting extra copies of it . . . Rudolf Kuhlmann, writing from Argentina, describes **apartment building air conditioning** there, where building is booming . . . What happens during a **range hood duct fire** and how the fire may be put out despite the excitement is the subject of a brief, practical article by H. G. Schaefer this month . . . Walter A. Becker describes the rebuilding of the Chicago stock yards—particularly the **International Amphitheater**—following last May's fire in his article in this issue . . . Many other articles, the regular departments, and the Journal Section of the A.S.H.V.E. complete the issue.

PREVIOUS predictions that 1935 would be an **air conditioning year** are substantiated by the installations made in Chicago for the first two months of 1935 as compared with the first two months of 1934 . . . In January 1934, six jobs were listed with a total horsepower of 527 and in February 1934 there were seven jobs with a total horsepower of 81¼. In January 1935 there were 13 jobs with 397½ horsepower, and in February 1935, 21 jobs with 1046 horsepower.

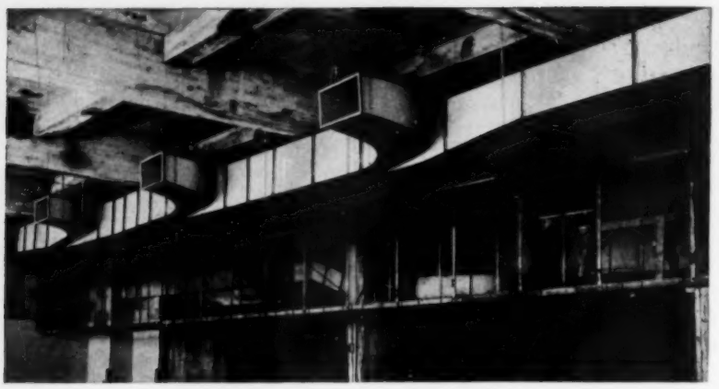
SCIENCE is resourceful," says the Montreal *Star*. "It couldn't pry open the Pullman windows so it air conditioned the train!" . . . But now that science has applied air conditioning to railway cars, it would find little use for a window-opener, were such a device to be invented . . . Conditioned air is better than outdoor air, usually a great deal cleaner.

RODOLFO Elias Calles, Governor of the Mexican State of Sonora, became interested in air conditioning through reading about it in various periodicals, now enjoys its benefits in his air conditioned private office, according to York Ice Machinery Corp., maker of the conditioning equipment . . . The Maytag hotel in Iowa will be one of the first hotels to be completely air conditioned, states the Frigidaire Corp., manufacturer of the equipment which is now being installed . . . Engineers of the Trane Co. have designed a cooling system to protect the cargoes of fruit (mostly bananas) carried by ships of the Standard Fruit & Steam Ship Co. . . . Cooling coils 18 ft long, about 5 ft high, are employed, and the water circulated through the coils is cooled by a steam jet refrigeration system . . . Summer air conditioning is first installed in the corner room where the boss sits, according to an executive who recently analyzed sales of conditioning equipment, then in various private offices, and finally the general office space is conditioned . . . From top to bottom the course of air conditioning wends its way . . . But at that, air conditioning was used to keep loaves of bread, printed paper, cigars, cigarettes, and scores of other products in shape before the boss gave it a chance to keep his own person in condition.

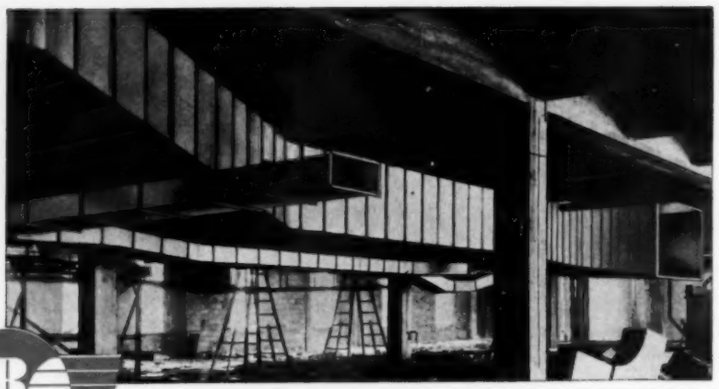
C. B. NOLTE was elected president of Crane Co. last month, and J. B. Berryman has become chairman of the board . . . Mr. Nolte comes to Crane from Robert W. Hunt Co., consulting, testing, and inspecting engineers, where he was successively division manager, manager, vice-president and general manager, and (in 1930) president and general manager, which position he held until elected to his present office.

BEING interested in welding generally and pipe welding in particular I was very much impressed by the article written by Sabin Crocker published in the February issue," writes a subscriber, "and have been trying ever since to find out more about the welding sleeves described, but without success; so by this letter I am asking you for the information" . . . Readers of H.P.&A.C. need have no hesitancy in writing for information to supplement our articles, or other heating, piping, or air conditioning data. **Dozens of requests** are handled by our editorial department every month and the opportunity to be of service to readers is welcomed.

THE Special Libraries Association is to hold its annual convention in Boston, June 11 to 14 . . . Over 1600 librarians who manage the libraries of industrial concerns, research laboratories, banks, offices, newspapers, museums, law, medical, scientific, engineering and other societies, as well as specialized departments in public, college and university libraries, comprise the membership . . . The librarian holds **a key position** in modern industry and commerce in making available to scientists, engineers, business men and others information from technical, engineering, scientific, and business periodicals, books and other publications . . . The activities of the librarians' association affects not only its membership, but all who depend on them for the information used in the day's work.



Republic U-LOY *ducts ventilate New York's U.S. Post Office Annex*



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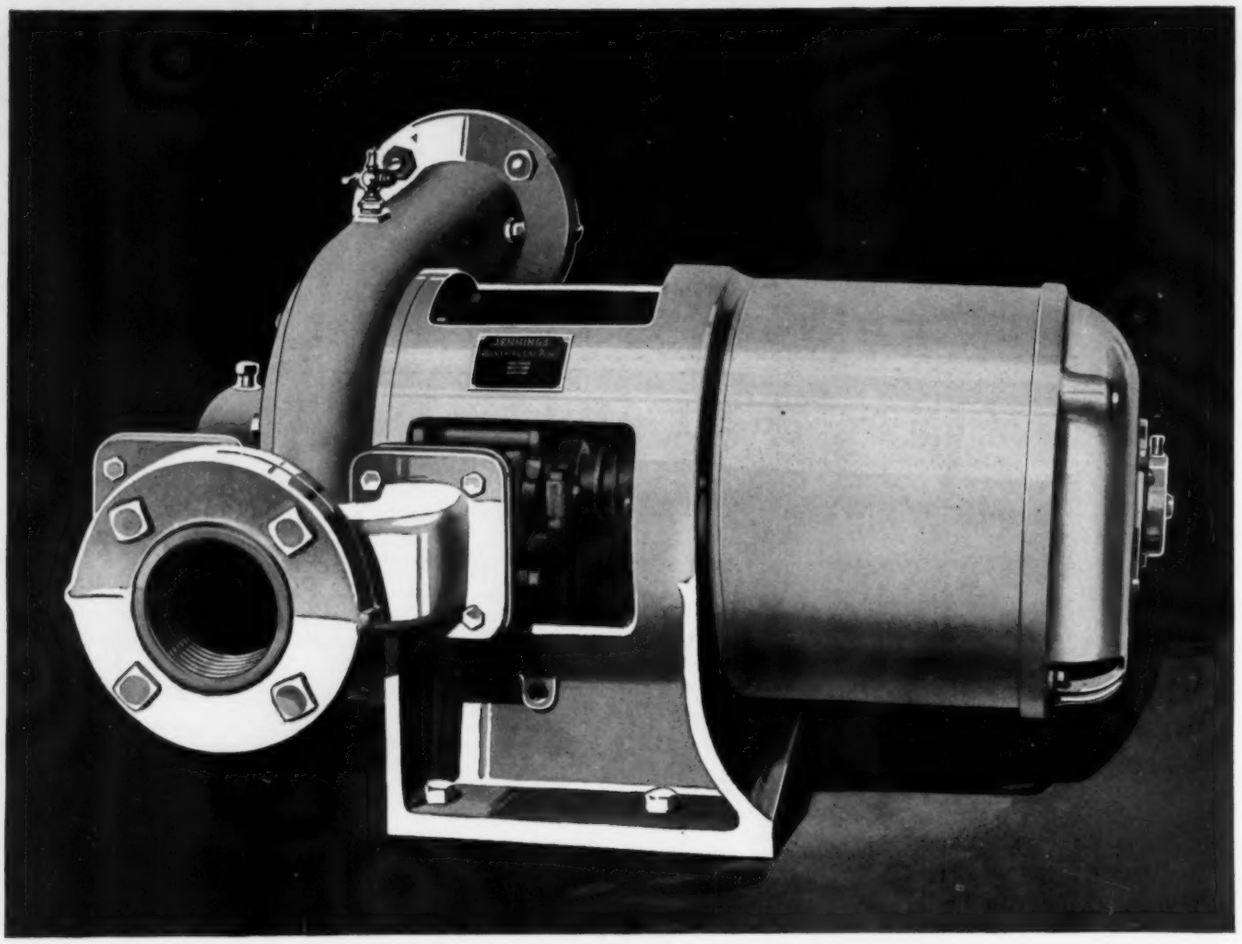

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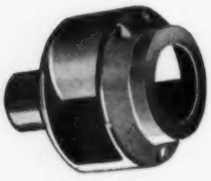

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Heating·Piping and Air Conditioning

April, 1935 • *Vol. 7, No. 4*

CHECK

Plant and Building Heating *NOW*

By **T. Napier Adlam***

THE end of the heating season is the time to review heating troubles and inefficiencies of the past winter and take steps to correct them. The heating equipment should be put in proper condition now for economical, efficient service next winter, rather than to put the job off until it is too late to do it justice. If the operating costs of a plant, office building, apartment, institution or other large building are analyzed, it will be found that the cost of steam and heat —in one of its forms—is probably the largest single item of expenditure. The question of heating economy therefore deserves very serious consideration—more consideration than often is given it. The object of this article is to suggest some of the ways in which this economy may be obtained. Because each plant and building has its own particular problems, it would be of course impossible to cover the entire subject. However a somewhat general review should be of aid in assuring that no possible economy be overlooked, and is particularly timely at this season.

Broadly speaking, there are four fundamental subdivisions of a heating system for a plant or building, each of which should be given proper consideration depending upon the way it functioned during the past heating season. These are (1) the heat generating plant; (2), the means of conveying the heat to the points of use; (3) the utilization of the heat; and (4) return of condensate (with a steam plant) to the heat generating apparatus.

* Chief Engineer, Sarco Co., Inc., New York, N. Y.

●

A check up of plant and building heating systems at the close of the heating season makes possible putting equipment in shape for better and more economical operation next winter. A check up now while the experiences of the past season are fresh in mind simplifies the job, assures ample time to make necessary changes, repairs, replacements . . . This article suggests some of the common faults to look for, is intended to supplement the operator's experience with his own system.

Watch Cost of Boiler Plant

Every boiler plant of reasonable size should be equipped with instruments and gages to show the amount of steam generated and the amount of fuel used, and such records should be analyzed weekly so that the actual cost of generating steam can be tabulated and examined continually.

Having had under my care the efficiency of a large number of steam generating plants (from steam heating systems to large power plants), it was necessary to tabulate and give monthly reports on all detail expenses. From these records I realized how easy it was for the efficiency to drop 15 to 20 per cent in a week, without any apparent indication to the operators. Hence there is very real reason for keeping a close check-up on all quantities and temperatures.

In addition to instruments for evaporation, gages should be provided for indicating draft conditions both at the chimney base and above the fire, and the temperature of the gases leaving the boiler should be known. The latter will indicate when the heating surface of the boiler is dirty.

Some of the salient points to watch for waste are:

(a) Inferior quality fuel.
(b) Short weight fuel supplied.
(c) Bad draft conditions. Too much or too little.
(d) Inefficiency of the oil burner.
(e) Leakage of air into the boiler flues.
(f) Dirty water in the boiler, with priming.
(g) Dirty boiler heating surfaces.

Fuel: Without instruments it is impossible to check up on (a) or (b), so that fuel may be delivered of inferior quality or with a large percentage of moisture. It is always advisable to check the weight of fuel delivered for any purpose, and for steam raising no safer way can be provided than instruments installed to check the total evaporation against the weight of fuel consumed. Short weight and inferior quality would show itself by low evaporation.

Draft: Every boiler should have a draft gage connected to the combustion chamber above the fire. Often it is advisable to have more than one gage on large boilers where it becomes necessary to check-up on the chimney and ashpit conditions, but a gage for draft above the fire is most essential. On one job in New York where an ordinary steam heating boiler was con-

nected to a fairly high chimney, giving about 0.75 in. W.G., one gage was purchased and connected to the chamber above the fire. The main draft damper was checked so that the draft above the fire was always kept down to 0.15 or 0.20 in. and this alone resulted in a saving of 2½ to 3 tons of coal per week. It paid for itself in about a month.

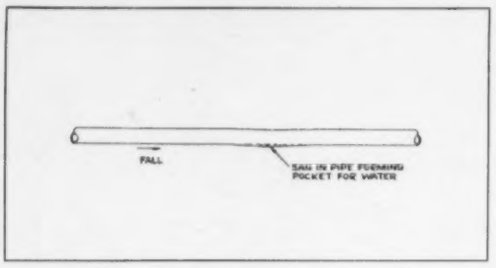

Fig. 1—Horizontal steam main with pocket causes water hammer. Such conditions should be corrected now

On a much larger job with which I was concerned, there were three steam boilers (100 lb steam pressure) each giving 8,000 lb of steam per hr; the draft was very poor and only expensive best quality fuel could be used to maintain the steam. A new chimney was out of the question. A special furnace for each boiler (total cost, $2,700) was recommended. Result: Ability to burn a much cheaper fuel and maintain the steam supply with ease, and a yearly saving of $6,000. A similar instance, somewhat larger than the above, suffering because of poor draft, was met by building a new chimney with a saving of 28 per cent in fuel consumption.

It is not always, however, that we want high draft conditions (as was seen by the first example) but a gage will soon pay for itself, because every boiler requires a definite draft condition for maximum efficiency, and this can only be ascertained and maintained by using gages.

Oil Burner: With an oil burning installation, checking up on evaporation is perhaps even more important, because an oil burner may easily drop in efficiency 30 to 40 per cent if not kept in good order. A check on the oil delivered should also be maintained.

The question of draft and atomization are among the most important factors in maintaining efficiency. With oil burners fitted with automatic control to start and stop according to pressure, as soon as the burner is switched off, cold air can be drawn into the combustion chamber and through the boiler passages, cooling the boiler and reducing its efficiency. In mild weather, when the burner would be idle for long periods, the efficiency drops and the saving that should be accomplished is partly nullified by the cold air passing through the boiler. Hence the need of a balanced draft damper.

Burners should be thoroughly examined and cleaned as soon as the heating season is over. Although using oil, there are invariably some parts which need oiling.

Air Leaks in Flues: One of the essentials of efficiency is to have an air tight flue and boiler. This applies to any kind of fuel whether gas, oil or coal. Cold air entering through badly fitted doors, plates or brickwork may increase fuel consumption 10 per cent higher than it should be. Test all joints with a candle to see if the flame is drawn inwards; if so, it should be corrected at once. Watch the temperature of the flue gases.

Boiler Water: The condition and level of the water in the boiler are very important to the satisfactory and efficient working of the system. With large steam generating plants a continuous blowdown with heat exchanger is advisable. The boiler water should be tested daily for density, alkalinity and causticity. Even with smaller plants it is very important to look after the condition of the water in the boiler because dirty water causes foaming and priming, and priming means wet steam, inefficient heating, and waste of fuel. Water floods the riser drains and steam cannot reach the extreme ends of the system, while the near sections are often overheated.

Too high water level in the boiler will also cause priming, which means noisy and unequal heating.

Two or more boilers coupled and working together will sometimes give different water levels. The remedy is to empty and clean out the boilers and then check the draft of the boiler having the lower water level until they balance.

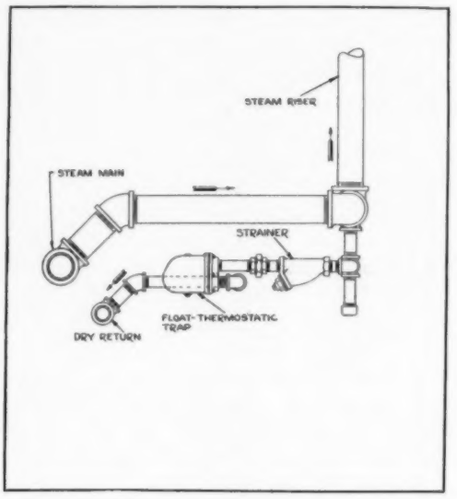

Fig. 2—Installation of combination traps

Incorrect application of boiler compound or other incorrect treatment of boiler feedwater means a big drop in efficiency and gives rise to a lot of trouble with pitting of boiler, corrosion of pipes, steam traps and valves. Blow down plenty. Test the density often.

Pitting of steam boilers often can be remedied by the use of zinc plates. Zinc plates about 1 in. thick, 6 in. wide and 12 in. long, suspended in the water area, with the addition of just sufficient soda ash to keep the water on the soft side, will often give excellent results.

Boiler Heating Surfaces: With any kind of boiler it is necessary to keep the heating surfaces well cleaned, and all direct heating and flue surfaces should be scraped in addition to brushing. I have increased the efficiency a further 8 per cent by simply scraping the surfaces after they have been thoroughly brushed in the usual way. Too much care cannot be taken of the boil-

ers, and where they are shut down for the summer the draft dampers, fire and flue doors should be left wide open to allow a constant current of air to pass through.

It is inevitable that a certain amount of sulphur will be deposited on fire and flue surfaces, and moisture will collect in the summer if a good current of air is not maintained. Moisture with the sulphurous deposit will eat away the metal and cause scaling quicker than anything else. Greater corrosion is caused if half burned fuel is left in the firebox.

Fire bars should be removed, if possible, and supporting pockets and trunnions cleaned to allow easy expansion. If this is not done, ash collected in the pockets will set up rust growth and fix the bars tightly in position, resulting in the sides being gradually pushed out, or other strains and damage to the boiler. Heating surfaces should be cleaned and painted.

With a hot water heating system, the boiler should be similarly treated, but the apparatus should not be left empty. It is advisable, at the end of the season, to run off just sufficient water at the draw-off valve on boiler until clear water appears; then close it up and leave it

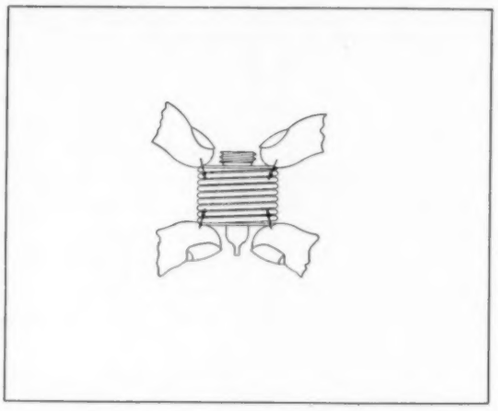

Fig. 3—A method of testing elements with pressure of thumb and forefinger. If corrugations move closer together element may be faulty.

until the heating season starts again. The air taps on all hot water radiators should be tested and a small amount of water drawn from each.

The Piping

The requirements for conveying steam are (1) Correct grading of the piping. (2) Efficient draining and trapping. (3) Insulation.

If there is noise when heating up, it is fairly certain that it is due to bad grading, trapping, or both. Very often the supports of a pipe will move and allow the pipe to drop, causing a low pocket for collecting water. This may not be much, and during the day may be silent, but if cool water collects during the night this will certainly give rise to water hammer when starting up in the morning (see Fig 1). Then again a steam pipe may be graded properly but the steam trap may be unable adequately to deal with the large amount of condensation when starting up. This will also cause knocking. Make sure you have the correct type and make of steam trap.

To cite an example, 40 faulty traps for the heating mains of a large institution were replaced with new ones (combination float and thermostatic type) and no

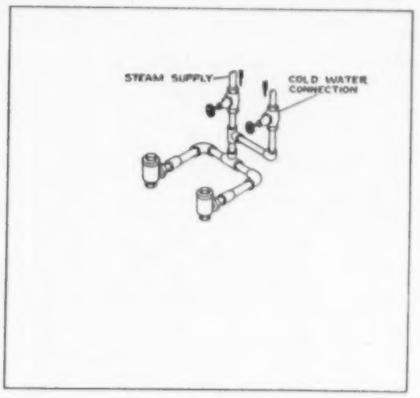

Fig. 4—Trap bodies set up in basement for testing elements

other changes were made. The result was a saving of 760 tons of coal annually at a cost of $1,000 for the traps. This meant that each faulty trap wasted equivalent to 19 tons of coal a year.

The uses of steam vary of course, depending upon the type of heating system, whether or not a laundry or kitchen is supplied, etc. With steam heating, with steam traps on the mains and on each of the radiators, every trap should be taken apart as soon as the heating season is over and the parts washed, including the inside of the body, with kerosene. This should be done with combination as well as the thermostatic traps. When working during the winter, the traps become coated with slime and often a caustic sediment, which will cause caustic embrittlement and destroy the elastic properties of the bellows. This causes the traps to leak and shortens their life.

When the thermostatic elements are taken out of the traps to be cleaned, each one should be examined to see if it is in good condition. One way is to hold the element between thumb and finger and press hard (see Fig. 3). If the corrugations can be compressed further together with the finger and thumb, the element may be dead, providing this test is made when the element is cold. There are some types of elements which cannot be tested this way but with those constructed with a high vacuum inside, this test is reliable.

A more practical test is to have a branch pipe connection in the steam piping in the basement, with one or more trap bodies fitted (Fig. 4). A dozen or so elements are removed from the radiator traps and tested in trap bodies in the basement to see if they close tight under steam pressure. Having selected the good ones, it is then possible to open up a similar number of other radiator trap bodies, wash out the bodies with kerosene and insert the tested and cleaned elements; removing the dirty elements to the basement for washing and testing, as before. In this way the whole system can be gone over with very little trouble. An alternative is first to buy a dozen or so new elements from the makers and start off by replacing with new ones the first batch of elements taken out, and then following the procedure suggested above throughout the system.

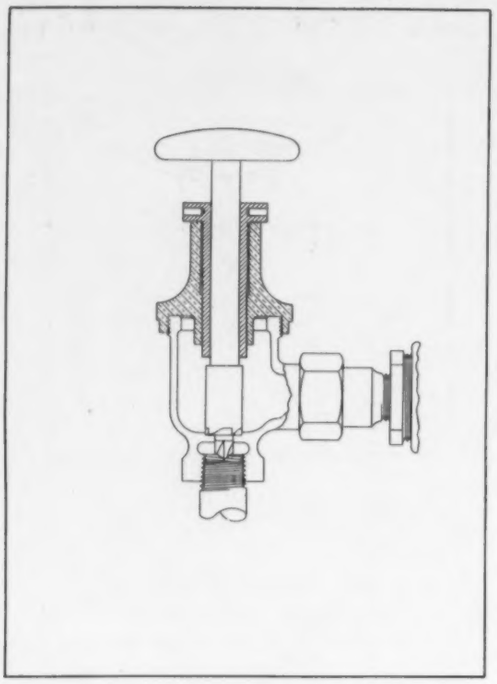

Fig. 5—Reaming tool to rectify faulty seats in old traps

Sometimes the valve seats in the trap bodies will be worn; if so, these should be fixed before replacing the elements. If renewable seats are fitted, it is easy to put in new seats. Where there are no renewable seats it is advisable to ream the seats in the trap bodies to form a perfect joint with the element head. Most trap makers supply reaming tools and special elements for reamed trap bodies (Fig. 5).

Another way to deal with old traps having no renewable seats is to tap out the hole in the trap body and supply a new seat with a new element. This makes practically a new trap which will last for years, and many firms can supply tools and units to make this change. (See Fig. 6.)

The annual cleaning and testing of traps has proved in many cases to give savings of 15 per cent and over in steam consumption, better heating, fewer complaints, and longer trap life. I have never known of the annual testing and cleaning procedure being discontinued once it was started.

[A second article will contain more suggestions on checking up heating systems at the end of the heating season.—The Editor.]

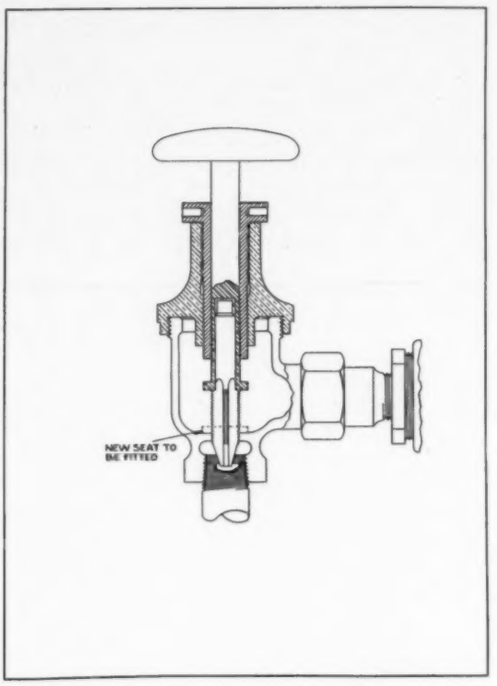

Fig. 6—Tapping tool to thread old trap body for renewable seat

Air Conditioning in Louisville

SUPPLEMENTING data for other cities published in the January, February, and March (1935) issues of H.P.&A.C., air conditioning installations in Louisville, Ky., for 1934 and previous to 1934 are reported here and are based on information furnished through the courtesy of W. D. Myers of the Louisville Gas and Electric Co. Prior to 1934 there were 19 jobs representing 1012.5 horsepower. Installations made during 1934 total 31, 7 of which were offices and 11 residences; the 31 represent 318.65 hp. Biggest of the 1934 jobs was a dry goods company accounting for 76.5 hp, while the 7 office installations account for a total of 14.2 hp and the 11 residences a total of 32.95 hp.

Thus, of the total of 50 jobs at the end of 1934, 31 or 62 per cent were installed that year. Of the total 1331.15 hp represented by the 50 installations, the 31 1934 jobs represent 318.65 hp or about 24 per cent. Assuming that 1.5 hp is equal to 1 ton, there is approximately 900 tons refrigeration capacity for air conditioning in Louisville.

Air Conditioning Installations in Louisville, Ky.

Year	Installation	Hp Con.	Kw Con.	Kw-Hr Per Season	Kw Addt'l Billing Demand
1926	Standard Gravure	15	11.25	5,625	8
1928	Loew's Theater	340	255	228,100	200
1929	American Tobacco Co.	280	210	478,500	120
1930	Brown Theater	50	37.5	40,000	20
1932	Will Sales	5	3.75	3,000	3
	Alamo Theater	30	22.5	26,207	15
	Strand Theater	5	3.75	18,134	3
	Kentucky Theater	26	19.5	21,640	12
1933	Brown Hotel Coffee Shop	8	6	6,000	4
	Kentucky Hotel	5.5	4	4,000	4
	Blue Boar Cafeteria	22.5	17	25,236	13
	Taylor Drugs, 4th & Chest.	10.5	7.75	13,470	6
	Reed Air Filter Co.	9.5	7.2	5,600	6
	Horney's Egg House	7	5.25	3,302	4
	S. & M. Cafeteria	21.5	16	31,270	12
	Ben Snyder	25	18.75	28,250	22
	Rialto Theater	145	108.75	117,480	100
	A. H. Bowman	3	2.25	425	2
	Merchants Ice & C. S.	4	3	3,404	3
1934	Taylor Drugs, 4th & Oak	6.5	5	9,000	4
	Lou. Taxicab & Trans. Co.	13	9.75	15,075	7
	Lyric Theater	30	22.5	17,980	16
	Jutt's Bar	5	3.75	3,700	3
	French Village	20	15	15,000	11
	Kresge's 5 & 10 Store	25	18.25	22,750	14
	Canary Cottage	15	11.2	19,130	8
	Byck's	23	16.5	11,530	12
	Stewart Dry Goods Co.	76.5	57.3	42,320	58
	WAVE Radio Station	6	4.5	6,750	3
	Lee Cralle	22	16.5	5,650	12
	Kunz	6.5	5	5,000	3
	Brown Hotel Grill	23	17.25	16,500	12
	(7) Offices	14.2	10.6	5,300	—
	(11) Residences	32.95	24.72	20,774	—

DESIGN OF MODERN INDUSTRIAL PIPING SYSTEMS —— Flexibility of Piping

It is the author's belief that too little consideration has been given to the inherent flexibility of pipe structures, where it is necessary to care for the deflection caused by thermal expansion. He reports here a method for the design of expansion bends using straight pipe and welding fittings, presents cases and tables aimed to simplify design . . . This information is the result of investigations and tests by the author, and has been used by him in the design of many industrial piping systems.

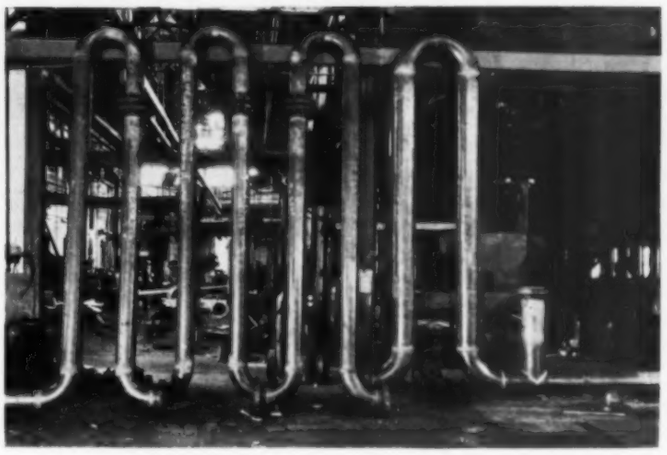

Fig. 1—Hair pin expansion bend for 500 lb, 750 F steam line

By **F. L. Snyder***

THE author does not attempt in this article to establish new theories that would require untold time to absorb and execute, but rather a safe and practical method for the design of expansion bends fabricated with welding fittings and straight pipe. To follow this plan, examples in Figs. 7-11 are arranged in a manner that will assist to make the design a simple problem. It will be noted in these examples that the method used would ordinarily be considered approximate, but upon check has been found to be consistently accurate and well within safe practical limits. The welding fittings are assumed to be free of bending or casting strains.

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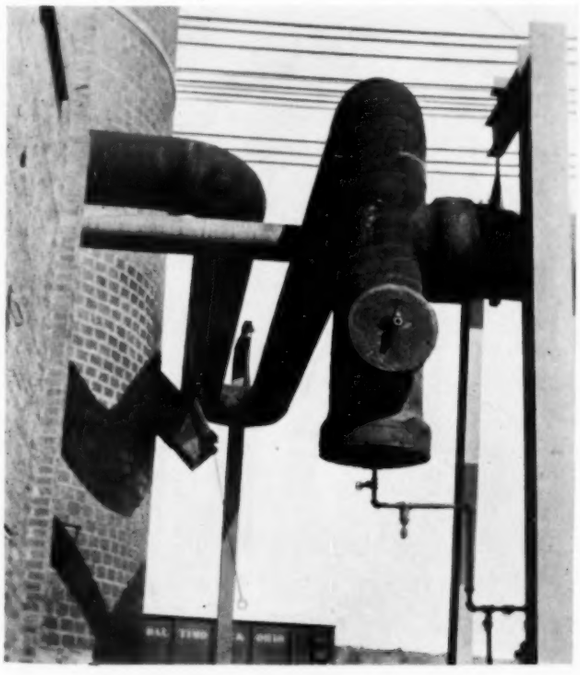


Fig. 2—Hair pin bend for 20 in. O.D. low pressure steam piping

Fig. 1 shows hair pin expansion bends that are being used in a 500 lb, 750 F steam line. These bends were designed to care for the entire deflection in the piping and are fabricated with straight pipe and welding fittings, the joints being electrically welded. Nominal diameter of the pipe is 8 and 10 in. and the wall thickness of both welding fittings and pipe is in accordance with the *A. S. T. M.* specifications for 600 lb S. W. P. The welding fittings have a radius of 1.5 pipe diameters.

The inherent flexibility of the welding fittings played an important part in the design of these bends since they reduced considerably the length of straight pipe that would otherwise have been used. Incidentally, they fitted in the limited space available, which naturally reduced the cost. Another and more important reason for selecting this type of expansion bend was that all physical values of both the pipe and fittings are known. No initial stresses or casting strains had to be considered.

Fig. 2 shows a hair pin bend used in a 20 in. O.D. low pressure steam line whose linear length between anchors is 400 ft. Welding fittings and straight pipe are the component parts. In this case the joints are gas welded. Consideration should be given to the minimum amount of space required for this type of structure, as well as the simplicity of design and fabrication.

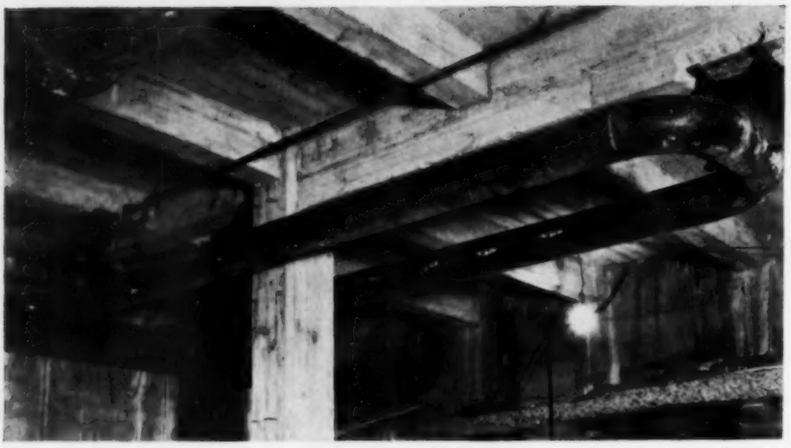


Fig. 3—Square expansion bend, used because it fitted perfectly with the structural arrangement of the building

Fig. 3 shows a square expansion bend. This particular type was used because it fitted perfectly with the structural arrangement of the building. It provides for the deflection of a 150 lb steam line which has a distance of 200 ft between anchors. The joints in the bend and the entire line are the gas welded butt type.

The preference for the hair pin or square expansion bend is dependent on local conditions. In many cases the hair pin is necessary due to the lack of available space, while the square bend, if space permits, may be designed to care for many times the deflection permissible in the hair pin. Another point in favor of the square bend is that it usually fits perfectly with the profile of the pipe line, especially so in long runs of outside pipe lines where it is necessary to cross roads, railroads, buildings, etc. When the square type of expansion bend is used in the upright position as shown in Fig. 4 care should be used in the selection of hangers. In the case where component parts are designed to absorb thermal expansion, the length L_3 shown on Fig. 8 should be supported with a spring hanger designed to take the dead load of the pipe in the deflected position. This relieves the length L_1 of stresses that would be necessary to overcome the dead load reactions if the pipe were supported with conventional hangers.

Figs. 1-4 are shown merely to convey the simplicity and practicability of the materials of construction now available to the designers of pipe structures.

Deflection Tests Made

During the past two years deflection tests were made with hair pin expansion bends, and also with straight pipe having welding fittings joined to the

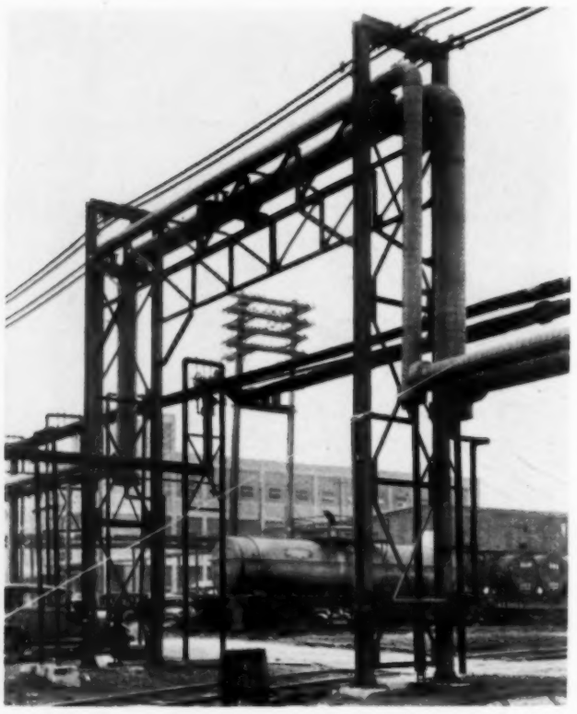


Fig. 4—Square expansion bend which serves a double purpose in carrying the pipe line over railroad track at industrial plant

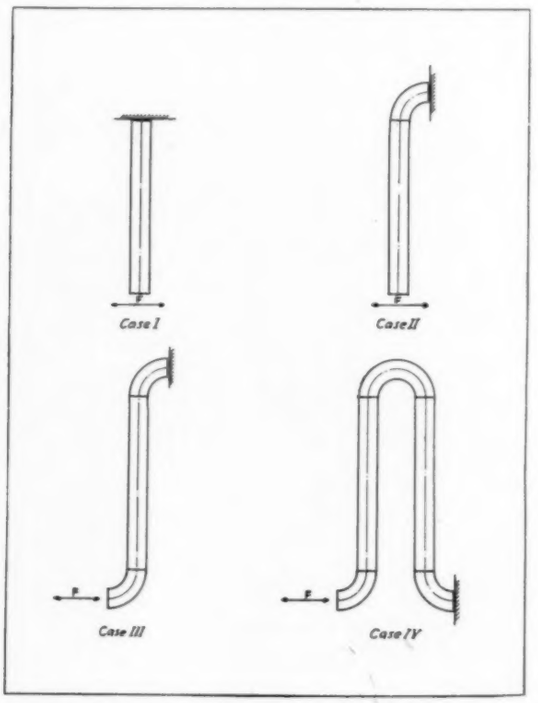


Fig. 5—The four units tested by the author

ends. The units tested are described in Fig. 5, Cases I, II, III, and IV. The object of these tests was to determine the flexibility provided by the welding fittings when used in the manner shown. It was found that a piece of straight pipe with a welding fitting joined to one end and anchored as shown in Case II has a greater deflection with the same force applied in direction F than a straight piece of pipe without the welding fitting and anchored as shown in Case I. It was also found that in each case the stresses developed by the acting force were the same. This indicated that the fundamental cantilever beam formula, corrected with a flexibility multiplication factor, in the case where welding fittings are used, is substantially correct and can be used with an assurance of accuracy when designing this type of expansion bend. The flexibility multiplication factor is used to correct the cantilever beam formula for the increased deflection apparent in the straight pipe when a welding fitting is joined to the end. In constructing the flexibility multiplication factor, which hereinafter will be referred to as f, three variables were considered; the radius of the neutral axis of the welding fitting; the mean radius of its diameter; and the moment of inertia of its transvere section. The above mentioned tests definitely proved that the flexibility of the fitting varied as these factors. The value $(1.5D)$ in f represents the radius of the welding fitting along its neutral axis, the numerical value of which should be changed to correspond to the radius of the fitting if a fitting having a radius other than 1.5 pipe diameters is used.

The deflection formula used to check the test Cases I, II, III, and IV is as follows:

CASE I

$$\Delta = SL^2 / 3Ey = L^3F / 3EI$$
$$S = 3Ey\Delta / L^2$$
$$L = \sqrt[3]{3EI\Delta / F} = \sqrt[3]{3Ey\Delta / S}$$
$$F = 3EI\Delta / L^3$$

CASE II

$$\Delta = SL^2f / 3Ey = L^3Ff / 3EI$$
$$S = 3Ey\Delta / L^2f$$
$$L = \sqrt[3]{3EI\Delta / Ff} = \sqrt[3]{3Ey\Delta / Sf}$$
$$F = 3EI\Delta / L^3f$$

CASE III

$$\Delta = 1.333SL^2f / 3Ey = SL^2f / 2.25Ey = FL^3f / 2.25EI$$
$$S = 2.25Ey\Delta / L^2f$$
$$L = \sqrt[3]{2.25EI\Delta / Ff} = \sqrt[3]{2.25Ey\Delta / Sf}$$
$$F = 2.25EI\Delta / L^3f$$

CASE IV

$$\Delta = 2SL^2f / 3Ey = SL^2f / 1.5Ey = FL^3f / 1.5EI$$
$$S = 1.5Ey\Delta / L^2f$$
$$L = \sqrt[3]{1.5EI\Delta / Ff} = \sqrt[3]{1.5Ey\Delta / Sf}$$
$$F = 1.5EI\Delta / L^3f$$

in which:

Δ = Deflection of pipe due to thermal expansion, inches.
F = Force required to deflect the pipe Δ inches.
E = Modulus of elasticity at working temperature.
S = Fibre stress, lb per sq in., at working temperature.
I = Moment of inertia = $0.049\ (d^4 - D_1^4)$.
f*= Flexibility multiplication factor $= (K + 10 / 5)\ (\sqrt[3]{K / 3.1})$.
K*= Flexibility coefficient $= r^2\ (1.5D)^{1.66} / 5.5I$.
r = Mean radius of pipe section in inches $= d - t / 2$.
t = Wall thickness of pipe, inches.
D = Nominal diameter of pipe, inches.
D_1 = Inside diameter of pipe, inches.
d = Outside diameter of pipe, inches.
y = Distance from farthest fibre to neutral axis of pipe, inches.

*For welding fittings having a radius of 1.5 pipe diameters—see text.

Table 1—Case II—Cantilever Beam Bend
One length of pipe 100" long and one 90° welding fitting

6" F.W.S. PIPE					
FORCE IN POUNDS	1000	2000	3000	4000	5000
DEFLECTION IN INCHES					
Test Average	1.009	2.03	2.855		
Calculated	0.915	1.83	2.745		
% Diff.	9.3	9.86	3.85		
FIBRE STRESS, lb per sq. in.					
Test Average	9800	19000	26000		
Calculated	11775	23500	35320		
% Diff.	16.77	19.15	26.4		
8" F.W.S. PIPE					
DEFLECTION IN INCHES					
Test Average	0.39	0.80	1.31	1.55	2.17
Calculated	0.373	0.746	1.119	1.492	1.865
% Diff.	4.36	6.75	14.575	3.74	14.05
FIBRE STRESS, lb per sq. in.					
Test Average	6100	10500	15400	22000	28500
Calculated	5940	11880	17820	23760	29700
% Diff.	2.62	11.6	13.6	7.4	4.09
10" F.W.S. PIPE					
DEFLECTION IN INCHES					
Test Average	0.22	0.3755	0.61	0.83	1.02
Calculated	0.171	0.342	0.511	0.684	0.855
% Diff.	22.3	8.92	16.22	17.6	16.17
FIBRE STRESS, lb per sq. in.					
Test Average	3400	6300	9000	12000	14900
Calculated	3350	6700	10050	14400	16750
% Diff.	1.47	5.97	10.45	16.66	11.04

Table 2—Case III—Cantilever Beam Bend
One length of pipe 100" long and two 90° welding fittings

6" F.W.S. PIPE					
FORCE IN POUNDS	1000	2000	3000	4000	5000
DEFLECTION IN INCHES					
Test Average	1.27	2.49	3.77		
Calculated	1.22	2.44	3.66		
% Diff.	3.94	2.01	2.92		
FIBRE STRESS, lb per sq. in.					
Test Average	10000	21000	27000		
Calculated	11770	23540	35310		
% Diff.	15.03	10.78	23.5		
8" F.W.S. PIPE					
DEFLECTION IN INCHES					
Test Average	0.615	1.09	1.61	2.21	2.7
Calculated	0.498	0.996	1.494	1.992	2.49
% Diff.	10.03	8.62	7.2	9.87	7.78
FIBRE STRESS, lb per sq. in.					
Test Average	5800	11000	17000	20000	25000
Calculated	5950	11900	17850	23800	29750
% Diff.	2.52	7.56	4.76	15.95	15.95
10" F.W.S. PIPE					
DEFLECTION IN INCHES					
Test Average	0.23	0.541	0.805	1.05	1.401
Calculated	0.2278	0.4556	0.6834	0.9112	1.139
% Diff.	0.957	15.775	15.1	13.22	18.7
FIBRE STRESS, lb per sq. in.					
Test Average	3000	6300	9000	12000	16000
Calculated	3340	6680	10020	13360	16700
% Diff.	10.17	5.69	10.18	10.17	4.19

Table 3—Case IV—Hair Pin Expansion Bend
Two lengths pipe 100" long, one 180°, two 90° fittings

6" F.W.S. PIPE					
FORCE IN POUNDS	1000	2000	3000	4000	5000
DEFLECTION IN INCHES					
Test Average	1.87	3.93	6.01		
Calculated	1.83	3.66	5.49		
% Diff.	2.14	6.87	8.66		
FIBRE STRESS, lb per sq. in.					
Test Average	11000	21000	28000		
Calculated	11775	23550	35325		
% Diff.	6.6	10.8	20.7		
8" F.W.S. PIPE					
DEFLECTION IN INCHES					
Test Average	0.791	1.61	2.36	3.19	4.011
Calculated	0.747	1.494	2.241	2.988	3.735
% Diff.	5.56	7.2	5.04	6.34	6.88
FIBRE STRESS, lb per sq. in.					
Test Average	5500	10500	16000	22000	27500
Calculated	5940	11880	17820	23760	29700
% Diff.	7.4	11.6	10.2	7.4	7.4
10" F.W.S. PIPE					
DEFLECTION IN INCHES					
Test Average	0.38	0.78	1.17	1.52	1.96
Calculated	0.342	0.684	1.026	1.368	1.71
% Diff.	10	12.3	12.3	10	12.75
FIBRE STRESS, lb per sq. in.					
Test Average	3500	6600	9000	12000	16000
Calculated	3350	6700	10050	13400	16750
% Diff.	4.28	1.49	10.45	10.45	3.14

Table 4—Standard Steel Pipe
A.S.T.M. Spec. A-120-32T & A-53-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¼	1.050	0.113	0.03704	1.216	1.472	0.5250
1	1.315	0.133	0.08734	1.961	1.550	0.6575
1¼	1.660	0.140	0.1947	2.839	1.620	0.8300
1½	1.900	0.145	0.3099	3.843	1.755	0.9500
2	2.375	0.154	0.6657	6.195	1.982	1.1875
2½	2.875	0.203	1.530	8.972	1.865	1.4375
3	3.500	0.216	3.017	12.15	1.912	1.7500
4	4.500	0.237	7.233	19.58	2.080	2.2500
5	5.563	0.258	15.16	28.36	2.227	2.7815
6	6.625	0.280	28.14	38.38	2.242	3.3125
8	8.625	0.322	72.49	61.87	2.355	4.3125
10	10.75	0.365	160.7	89.60	2.390	5.375
12	12.75	0.375	279.3	121.3	2.572	6.375
14	14	0.375	372.8	156.7	2.900	7
16	16	0.375	562.1	195.5	3.092	8
18	18	0.375	806.6	237.7	3.281	9
20	20	0.375	1113.0	283.2	3.462	10
24	24	0.4375	2248.0	383.3	3.372	12

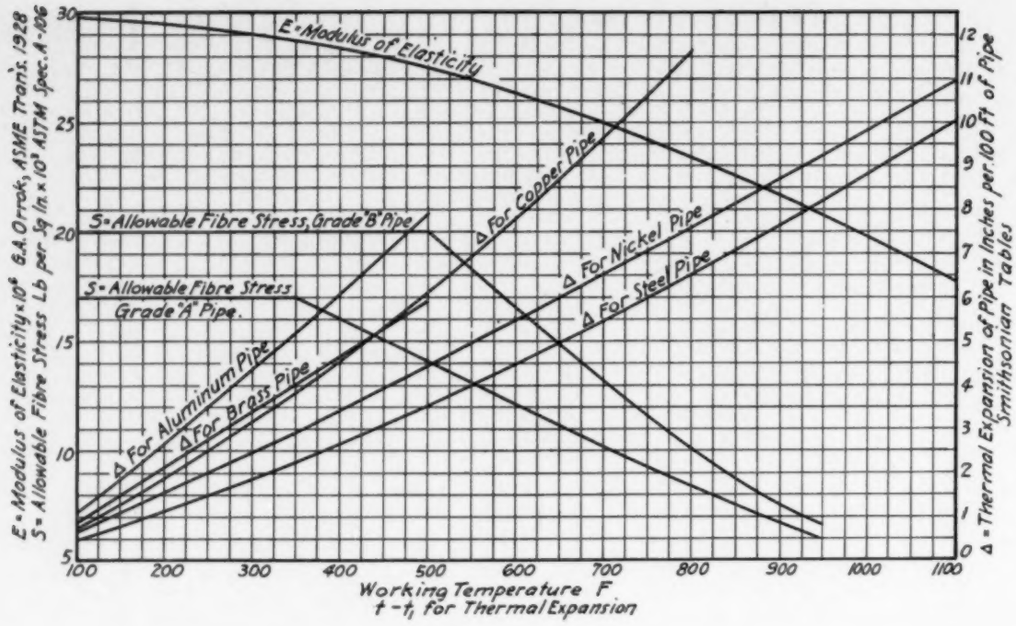

E = Modulus of Elasticity x 10⁶ G.A. Orrok, ASME Trans. 1928
S = Allowable Fibre Stress Lb per Sq In. x 10³ ASTM Spec. A-106
E = Modulus of Elasticity
S = Allowable Fibre Stress, Grade 'B' Pipe
S = Allowable Fibre Stress Grade 'A' Pipe.
Δ For Copper Pipe
Δ For Nickel Pipe
Δ For Steel Pipe
Δ For Aluminum Pipe
Δ For Brass Pipe
Δ = Thermal Expansion of Pipe in Inches per 100 Ft of Pipe Smithsonian Tables
Working Temperature F
t - t₁ for Thermal Expansion

Fig. 6—Modulus of elasticity, allowable fibre stress, and thermal expansion of pipes

It will be noted that in Cases III and IV the numerical constant has been changed. This is necessary in order to design the unit with one operation, rather than to determine the deflection in each length of pipe and fitting, which would be rather laborious and hardly practical when considering the small difference in the ultimate result. In both Cases III and IV, the full value of the flexibility provided by the welding fitting, when used as in Case II, cannot be taken advantage of because of the fact that the location of the neutral axis of the structural moment[1] in either unit changes, and high stresses are concentrated in the pipe walls. Therefore, it is also recommended that the maximum design stress, or the modulus of elasticity should not exceed the values shown in Fig. 6 at the working temperature.

The straight or connecting pipe to a hair pin or square expansion bend, or the connecting pipe to a direction change, is considered as a column, since it is supported at intervals and at distances between supports that will reduce dead load deflection to a minimum. This connecting pipe reacts the forces created by thermal expansion to the anchors, the expansion bends, and direction changes. Therefore, it is in compression through its entire transverse section, the value of which is usually so small per unit of area that it is regarded as a negligible quantity when considering the stresses in the pipe structure.

The flexibility of the welding fitting is directly credited to the flattening of its transverse section when forces are applied at right angles to its neutral axis, and varies as its nominal diameter and effective radius. The radius varies the linear length of the tube; a 6 in. fitting with a radius of 1.5 pipe diameters if straightened would have a linear length of 14.9 in., while a 24 in. fitting having the same radius

[1] "A Graphic Method for Determining Expansion Stresses in Pipe Lines," by C. T. Mitchell. *A.S.M.E. Trans.* Part II FSP-52-25.

Table 5—Extra Heavy Pipe Lap-Welded and Seamless Steel A.S.T.M. Spec. A-120-32T & A-53-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.154	0.04479	1.216	1.243	0.5250
1	1.315	0.179	0.1056	1.961	1.314	0.6575
1¼	1.660	0.191	0.2418	2.839	1.360	0.8300
1½	1.900	0.200	0.3912	3.843	1.456	0.9500
2	2.375	0.218	0.8679	6.195	1.606	1.1875
2½	2.875	0.276	1.924	8.972	1.554	1.4375
3	3.500	0.300	3.894	12.15	1.570	1.7500
4	4.500	0.337	9.610	19.58	1.671	2.2500
5	5.563	0.375	20.67	28.36	1.719	2.7815
6	6.625	0.432	40.49	38.38	1.703	3.3125
8	8.625	0.500	105.70	61.87	1.772	4.3125
10	10.75	0.500	212.00	89.6	1.940	5.375
12	12.75	0.500	361.50	121.3	2.110	6.375
14	14	0.593	561.54	156.7	2.106	7
16	16	0.656	930.51	195.5	2.088	8
18	18	0.718	1456.77	237.7	2.066	9
20	20	0.812	2254.00	283.2	1.992	10
24	24	0.937	4512.90	383.3	1.966	12

Table 6—250 Lb Std. Lap-Welded 750 F Working Temperature A.S.T.M. Spec. A-106-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.113	0.03704	1.216	1.472	0.5250
1	1.315	0.133	0.08734	1.961	1.550	0.6575
1¼	1.660	0.140	0.1947	2.839	1.620	0.8300
1½	1.900	0.145	0.3099	3.843	1.755	0.9500
2	2.375	0.154	0.6657	6.195	1.982	1.1875
2½	2.875	0.203	1.530	8.972	1.865	1.4375
3	3.500	0.216	3.017	12.15	1.912	1.7500
4	4.500	0.237	7.233	19.58	2.080	2.2500
5	5.563	0.258	15.16	28.36	2.227	2.7815
6	6.625	0.280	28.14	38.38	2.242	3.3125
8	8.625	0.322	72.49	61.87	2.355	4.3125
10	10.75	0.365	160.70	89.60	2.390	5.375
12	12.75	0.406	299.88	121.3	2.433	6.375
14	14	0.437	429.10	156.7	2.593	7
16	16	0.500	731.90	195.5	2.512	8
18	18	0.562	1172.00	237.7	2.437	9
20	20	0.593	1700.79	283.2	2.463	10
24	24	0.687	3424	383.3	2.412	12

Table 7—300 Lb Std. Seamless Steel, 750 F Working Temperature A.S.T.M. Spec. A-106-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.113	0.03704	1.216	1.472	0.5250
1	1.315	0.133	0.08734	1.961	1.550	0.6575
1¼	1.660	0.140	0.1947	2.839	1.620	0.8300
1½	1.900	0.145	0.3099	3.843	1.755	0.9500
2	2.375	0.154	0.6657	6.195	1.982	1.1875
2½	2.875	0.203	1.530	8.972	1.865	1.4375
3	3.500	0.216	3.017	12.15	1.912	1.7500
4	4.500	0.237	7.233	19.58	2.080	2.2500
5	5.563	0.258	15.16	28.36	2.227	2.7815
6	6.625	0.280	28.14	38.38	2.242	3.3125
8	8.625	0.277	63.35	61.87	2.616	4.3125
10	10.75	0.307	137.40	89.60	2.707	5.375
12	12.75	0.330	248.50	121.3	2.821	6.375
14	14	0.375	372.80	156.7	2.900	7
16	16	0.375	562.10	195.5	3.092	8
18	18	0.437	931.3	237.7	2.921	9
20	20	0.500	1457	283.2	2.784	10
24	24	0.562	2846	383.3	2.787	12

would have a linear length of 37.7 in., for instance. This is constant within the proportional limits of their respective moments of inertia. Therefore, the fitting is considered a cantilever beam as much as the straight connecting pipe. Sufficient data were compiled from tests on smaller installations to support this theory.

It was determined that the point of greatest moment in all of the tested units occurred some distance from the fitting in the connecting pipe. The point of greatest stress varied slightly with the nominal diameter of the pipe. With a measured stress of 15,000 per sq in. at this point, the stress measured in the 180 deg welding fitting, in the case of the hair pin bend, was 9500 lb per sq in., and in the 90 deg fitting it was 11,500 lb per sq in.

The flexibility multiplication factor f was constructed to vary with the radius of the welding fitting, its moment of inertia, and the mean radius of its transverse section, and to support the cantilever beam formula in checking the test results. The fibre stress created in the pipe walls is directly proportional to the force applied and since the force is

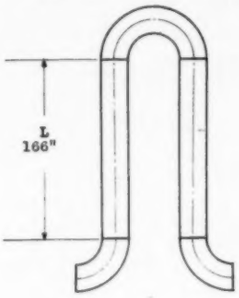


Fig. 7

Assuming an 8" standard weight steel pipe line transporting steam at 750 F. The thermal expansion is 5" and will be cared for with a "hair pin expansion bend" as shown in the diagram. Using Case IV formula and substituting the following values:

$\Delta = 5$
$I = 72.49$ from Table No.4
$f = 2.355$ from Table No.4
$y = 4.3125$ from Table No.4
$E = 25000000$ at 750 F working temperature from Chart on Fig. 6
$S = 12500$ at 750 F working temperature from Chart on Fig. 6

1 - With the formula determine the length L

$$L = \sqrt[2]{1.5Ey\Delta} / Sf = \sqrt[2]{1.5 \times 25000000 \times 4.3125 \times 5} / 12500 \times 2.355 = 166"$$

2 - Determine the force "F" reacted against the anchors thru the connecting pipe.

$$F = 1.5EI\Delta / L^3f = 1.5 \times 25000000 \times 72.49 \times 5 / 166^3 \times 2.355 = 1265\#$$

If the connecting pipe is cold sprung 5", this bend may be used to care for a total expansion of 10".

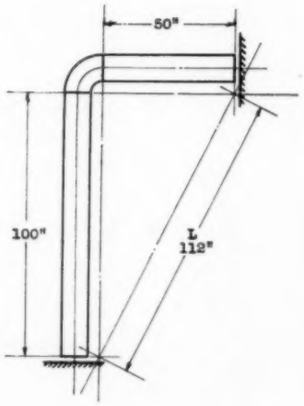


Fig. 9

Assuming an 8" pipe line transporting steam at 750 F between two points as shown in the diagram. Determine the available deflection in the pipe structure and the force acting against the anchor points, using Case II formula and substituting the following values:

$I = 72.49$ from Table No.4
$f = 2.355$ from Table No.4
$y = 4.3125$ from Table No.4
$E = 25000000$ at 750 F working temperature from chart on Fig. 6
$S = 12500$ at 750 F working temperature from chart on Fig. 6

1 - Determine length L by trigonometrical function

$$L = \sqrt[2]{100^2 + 50^2} = 112"$$

2 - Determine available deflection

$$\Delta = 2L^2f / 3Ey = 12500 \times 112^2 \times 2.355 / 3 \times 25000000 \times 4.3125 = 1.14"$$

3 - Determine force acting against the anchor points

$$F = 3EI\Delta / L^3f = 3 \times 25000000 \times 72.49 \times 1.14 / 112^3 \times 2.355 = 1870 \text{ lbs.}$$

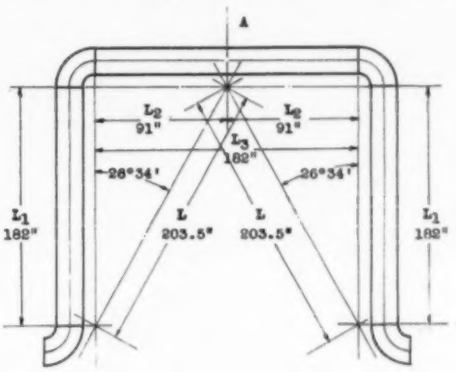


Fig. 8

Assuming an 8" standard weight steel pipe line transporting steam at 750 F. The total expansion is 10" and will be cared for with a "square expansion bend" as shown in the diagram above. The lengths L_1, and L_2 will do an equal amount of work. Therefore, the unit is theoretically divided at the line A, on the basis that each section will provide for one half of the total expansion or 5". This permits the use of Case III formula. Substitute the following values in the formula:

$\Delta = 5$
$I = 72.49$ from Table No.4
$f = 2.355$ from Table No.4
$y = 4.3125$ from Table No.4
$E = 25000000$ at 750 F working temperature from Chart on Fig. 6
$S = 12500$ at 750 F working temperature from Chart on Fig. 6

1 - With the formula determine length L

$$L = \sqrt[2]{2.25Ey\Delta} / Sf = \sqrt[2]{2.25 \times 25000000 \times 4.3125 \times 5} / 12500 \times 2.355 = 203.5$$

2 - Determine length L_2 by trigonometrical function, multiplying L with the sine of 26°34' or .44724. $L_2 = .44724 \times 203.5 = 91"$

3 - Determine length L_1 by trigonometrical function, multiplying L with the cosine of 26°34' or .89441. $L_1 = .89441 \times 203.5 = 182"$

If the connecting pipe is cold sprung 10", this bend may be used to care for a total expansion of 20".

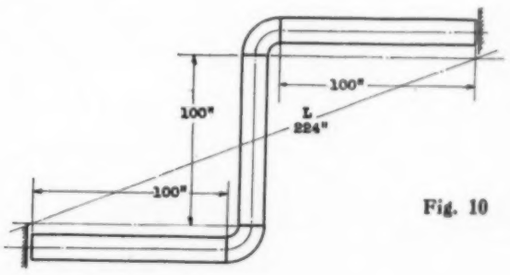


Fig. 10

Assuming an 8" pipe line transporting steam at 750 F between two points as shown in the diagram. Determine the available deflection in the pipe structure and the force acting against the anchor points, using Case III formula and substituting the following values:

$I = 72.49$ from Table No.4
$f = 2.355$ from Table No.4
$y = 4.3125$ from Table No.4
$E = 25000000$ at 750 F working temperature from chart on Fig. 6
$S = 12500$ at 750 F working temperature from chart on Fig. 6

1 - Determine length L by trigonometrical function

$$L = \sqrt[2]{200^2 + 100^2} = 224"$$

2 - Determine available deflection

$$\Delta = 3L^2f / 2.25Ey = 12500 \times 224^2 \times 2.355 / 2.25 \times 25000000 \times 4.3125 = 6.1"$$

3 - Determine force acting against anchor points

$$F = 2.25EI\Delta / L^3f = 2.25 \times 25000000 \times 72.49 \times 6.1 / 224^3 \times 2.355 = 935\#$$

directly proportional to the moment of inertia of the section, this factor will apply to pipe of any wall thickness, without the support of a rigidity or stress multiplication factor.

Tables 1, 2, and 3 compare the test results with the calculated results. Fig. 6 gives the modulus of elasticity at working temperature, the fibre stress in lb per sq in., and the thermal expansion of different pipe materials at the temperature difference $(t-t_1)$.

Tables 4, 5, 6, 7, 8, 9, 10, 11, and 12 give the values of D, d, t, I, $(1.5D)^{1.66}$, f, and y for all classes of standard steel pipe up to 1500 lb. All calculations in the tables and in the examples are subject to slide rule error.

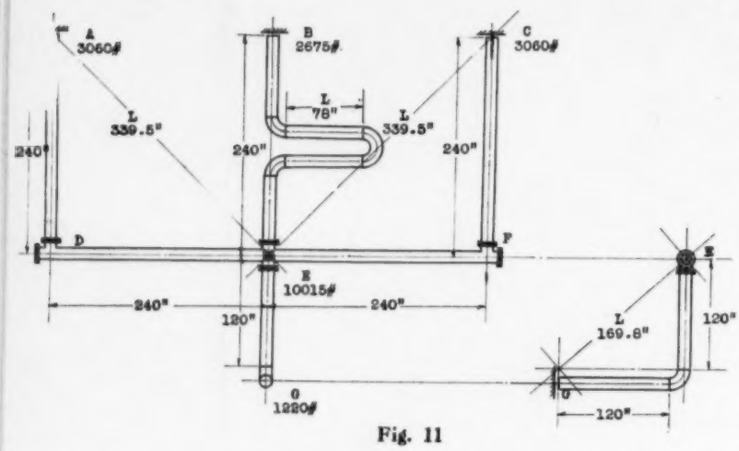


Fig. 11

Assuming a pipe structure anchored at points A, B, C, E, and G. The pipe is 8" standard weight steel and transports steam at 750 F. A "hair pin expansion bend" is used to care for the thermal expansion between points B and E. The inherent flexibility of the straight pipe is utilized between points E, D, and A, and E, F, and C. In this case, Case IV formula is used to determine the size of the "hair pin expansion bend", and Case I formula is used to determine the available flexibility between points E, D, and A, and E, F, and C, and Case II formula between points E, and G. Substituting the following values in the formulas:

I = 72.49 from Table No. 4
f = 2.355 from Table No. 4
y = 4.3125 from Table No. 4
E = 25000000 at 750 F working temperature from Chart on Fig. 6
S = 12500 at 750 F working temperature from Chart on Fig. 6

1 - Refer to chart on Fig. 6 to determine the amount of thermal expansion in the various units. The atmospheric temperature at the time of erection is 50 F, then $(t-t_1) = 750 - 50 = 700$ F. Referring to the 700 F line on the $(t - t_1)$ Scale, follow vertically to the point of intersection on the curve Δ FOR STEEL PIPE and then straight across to the right read 5.5", which is the thermal expansion per 100'00" of pipe. For 240" $\Delta = 240 \times 5.5 / 12 \times 100 = 1.1"$ For 120" $\Delta = 120 \times 5.5/12 \times 100 = .55"$

2 - Determine the length L between points E and A, and E and C and E and G, by trigonometrical function.

Points E and A, and E and C, $L = \sqrt[2]{240^2 + 240^2} = 339.5"$

Points E and G, $L = \sqrt{120^2 + 120^2} = 169.8"$

3 - The amount of thermal expansion between points E and B as determined above is 1.1". Using Case IV formula the length L of the "hair pin expansion bend" is

$L = \sqrt[2]{1.5EyΔ / Sf} = \sqrt[2]{1.5 \times 25000000 \times 4.3125 \times 1.1 / 12500 \times 2.355} = 78"$

and the force acting against points E and B is

$F = 1.5EIΔ / L^3f = 1.5 \times 25000000 \times 72.49 \times 1.1 / 78^3 \times 2.355 = 2675\#$

4 - Determine the available deflection between points E and A, and E and C, where the length L = 339.5". Using Case I formula

$Δ = SL^2 / 3Ey = 12500 \times 339.5^2 / 3 \times 25000000 \times 4.3125 = 4.45"$
This is sufficient to care for the total expansion of 2.2"

and the force acting against points E and A, and E and C is (based on a total deflection of 2.2")

$F = 3EIΔ / L^3 = 3 \times 25000000 \times 72.49 \times 2.2 / 339.5^3 = 3060\#$

5 - Determine the available deflection between points E and G, where the length L = 169.8". Using Case II formula

$Δ = SL^2f / 3Ey = 12500 \times 169.8^2 \times 2.355 / 3 \times 25000000 \times 4.3125 = 2.622"$
This is sufficient to care for the total expansion of 1.1"

and the force acting against points E and G is (based on a total deflection of 1.1")

$F = 3EIΔ / L^3f = 3 \times 25000000 \times 72.49 \times 1.1 / 169.8^3 \times 2.355 = 520\#$

6 - The total force acting against the anchor point E, is the sum of the forces determined in 3, 4, and 5, or 2675 + 3060 + 520 = 9315#.

Table 8—400 Lb Std. Lap-Welded 750 F Working Temperature
A.S.T.M. Spec. A-106-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.154	0.04479	1.216	1.243	0.5250
1	1.315	0.179	0.1056	1.961	1.314	0.6575
1¼	1.660	0.191	0.2418	2.839	1.360	0.8300
1½	1.900	0.200	0.3912	3.843	1.456	0.9500
2	2.375	0.218	0.8679	6.195	1.606	1.1875
2½	2.875	0.276	1.924	8.972	1.554	1.4375
3	3.500	0.300	3.894	12.15	1.570	1.7500
4	4.500	0.337	9.61	19.58	1.671	2.2500
5	5.563	0.375	20.67	28.36	1.719	2.7815
6	6.625	0.432	40.49	38.38	1.703	3.3125
8	8.625	0.500	105.70	61.87	1.772	4.3125
10	10.75	0.500	212.00	89.60	1.940	5.375
12	12.75	0.562	399.84	121.3	1.960	6.375
14	14	0.593	561.54	156.7	2.106	7
16	16	0.656	930.51	195.5	2.083	8
18	18	0.718	1456.77	237.7	1.896	9
20	20	0.812	2254.00	283.2	1.993	10
24	24	0.937	4512.90	383.3	1.963	12

Table 9—400 Lb Std. Seamless Steel, 750 F Working Temperature
A.S.T.M. Spec. A-106-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.113	0.03704	1.216	1.472	0.5250
1	1.315	0.133	0.08734	1.961	1.550	0.6575
1¼	1.660	0.140	0.1947	2.839	1.620	0.8300
1½	1.900	0.145	0.3099	3.843	1.755	0.9500
2	2.375	0.154	0.6657	6.195	1.982	1.1875
2½	2.875	0.203	1.530	8.972	1.865	1.4375
3	3.500	0.216	3.017	12.15	1.912	1.7500
4	4.500	0.237	7.233	19.58	2.080	2.2500
5	5.563	0.258	15.16	28.36	2.227	2.7815
6	6.625	0.280	28.14	38.38	2.242	3.3125
8	8.625	0.322	72.49	61.87	2.355	4.3125
10	10.75	0.365	160.70	89.60	2.390	5.375
12	12.75	0.406	299.88	121.3	2.433	6.375
14	14	0.437	429.10	156.7	2.593	7
16	16	0.500	731.90	195.5	2.512	8
18	18	0.562	1172.00	237.7	2.437	9
20	20	0.593	1700.79	283.2	2.463	10
24	24	0.687	3424	383.3	2.412	12

Table 10—600 Lb Std. Seamless Steel, 750 F Working Temperature
A.S.T.M. Spec. A-106-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.154	0.04479	1.216	1.243	0.5250
1	1.315	0.179	0.1056	1.961	1.314	0.6575
1¼	1.660	0.191	0.2418	2.839	1.360	0.8300
1½	1.900	0.200	0.3912	3.843	1.456	0.9500
2	2.375	0.218	0.8679	6.195	1.606	1.1875
2½	2.875	0.276	1.924	8.972	1.554	1.4375
3	3.500	0.300	3.894	12.15	1.570	1.7500
4	4.500	0.337	9.610	19.58	1.671	2.2500
5	5.563	0.375	20.67	28.36	1.719	2.7815
6	6.625	0.432	40.49	38.38	1.703	3.3125
8	8.625	0.500	105.70	61.87	1.772	4.3125
10	10.75	0.593	244.41	89.60	1.745	5.375
12	12.75	0.687	474.32	121.3	1.727	6.375
14	14	0.750	687.30	156.7	1.811	7
16	16	0.843	1153.95	195.5	1.779	8
18	18	0.937	1839.46	237.7	1.752	9
20	20	1.031	2762.13	283.2	1.718	10
24	24	1.218	5658.52	383.3	1.667	12

Table 11—900 Lb Std. Seamless Steel, 750 F Working Temperature
A.S.T.M. Spec. A-106-33T

D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.154	0.04479	1.216	1.243	0.5250
1	1.315	0.179	0.1056	1.961	1.314	0.6575
1¼	1.660	0.191	0.2418	2.839	1.360	0.8300
1½	1.900	0.200	0.3912	3.843	1.456	0.9500
2	2.375	0.218	0.8679	6.195	1.606	1.1875
2½	2.875	0.276	1.924	8.972	1.554	1.4375
3	3.500	0.300	3.894	12.15	1.570	1.7500
4	4.500	0.437	11.65	19.58	1.443	2.2500
5	5.563	0.500	25.69	28.36	1.455	2.7815
6	6.625	0.562	49.53	38.38	1.461	3.3125
8	8.625	0.593	121.12	61.87	1.600	4.3125
10	10.75	0.718	285.62	89.60	1.556	5.375
12	12.75	0.843	560.61	121.3	1.527	6.375
14	14	0.937	823.12	156.7	1.585	7
16	16	1.031	1362.00	195.5	1.577	8
18	18	1.156	2175.81	237.7	1.54	9
20	20	1.25	2981.00	283.2	1.6125	10
24	24	1.5	6727.45	383.3	1.4725	12

Table 12—1500 Lb Std. Seamless Steel, 750 F Working Temperature
A.S.T.M. Spec. A-106-33T

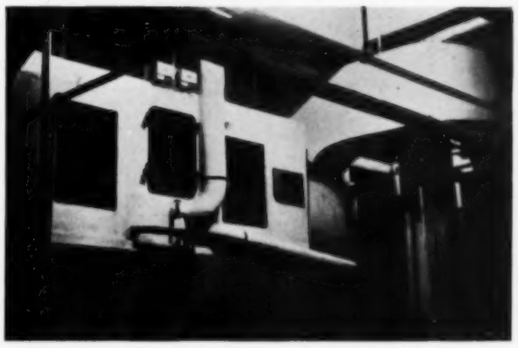
D	d	t	I	$1.5D^{1.66}$	f	y
¾	1.050	0.218	0.05260	1.216	1.040	0.5250
1	1.315	0.250	0.12492	1.961	1.106	0.6575
1¼	1.660	0.250	0.28330	2.839	1.180	0.8300
1½	1.900	0.281	0.48140	3.843	1.295	0.9500
2	2.375	0.343	1.16042	6.195	1.309	1.1875
2½	2.875	0.375	2.35270	8.972	1.310	1.4375
3	3.500	0.437	5.02495	12.15	1.271	1.7500
4	4.500	0.531	12.789	19.58	1.319	2.2500
5	5.563	0.625	29.972	28.36	1.285	2.7815
6	6.625	0.718	56.720	38.38	1.303	3.3125
8	8.625	0.906	165.600	61.87	1.260	4.3125
10	10.75	1.125	398.525	89.60	1.210	5.375
12	12.75	1.312	779.688	121.3	1.196	6.375
14	14	1.406	1114.652	156.7	1.260	7
16	16	1.562	1864.400	195.5	1.247	8
18	18	1.750	2978.075	237.7	1.219	9
20	20	1.937	4526.62	283.2	1.196	10
24	24	2.312	9352.63	383.3	1.157	12

Apartment Building Air Conditioning in Argentina

By **Rudolf Kuhlmann***

Fig. 1 (above)—The highest building in South America, nearing completion, providing apartment space for 120 families. . . Fig. 2 (below)—A view of some of the air conditioning equipment in a nine-story apartment building

ONE of the most interesting phases of the development of air conditioning in Buenos Aires, the metropolis of South America, is the introduction of air conditioning in the apartment house field. The Edificio Plaza San Martin (Kavanagh Building) shown in Fig. 1 is perhaps the most striking installation of this type. It has been under construction for the last year and should be completed soon; when finished it will be the tallest building in South America, with a total height of 33 stories, 29 of which will be used for apartment space for 120 families. Mechanical equipment will occupy the remaining four floors.



System in 33-Story Building Has Unusual Features

The air conditioning system is unique in some respects. A 400-ton centrifugal compressor is installed to provide the necessary refrigeration in connection with two washers. Condenser cooling water at 66 F is pumped from a well 150 ft deep. After the water has passed through the condensers, it is at the right temperature for a swimming pool, to which it is fed in a continuous flow to assure clean pool water at all times.

An unusual feature for an installation of this size is that no return air system is provided. The engineers who designed this job were pioneering a bit in this respect, for there is very little precedent on conditioning this kind of building. It seemed inadvisable to bring back the air from one apartment to the central system and distribute it to the rooms occupied by other families.

A very practical arrangement of air distribution is used. A typical apartment consists of a living room, dining room, 3 bedrooms and a kitchen. There is no supply grille in the kitchen, but the other rooms each have a separate supply. The ducts for the bedrooms are grouped into one large duct, as are those for the living and dining rooms; these two large ducts lead from the main supply duct, where a damper is installed so that the cooled air may be supplied alternately to one section of the apartment or the other, or partly to each. The damper is hand-controlled and is located in the kitchen ceiling.

It is felt that the living-dining room section will not need maximum cooling at the same time as the bedroom section. Additional control is provided through adjustments in the individual supply grilles.

Other Apartment Building Installations

A "pocket edition" of this same type of system has been installed in a smaller apartment building and has been operating during the present summer.[1] In this building are two similar but completely separate air conditioning systems. What appears to be one apartment building was actually built in two sections, so that either part could later be sold to different purchasers.

Each half of this building has nine stories, consisting of 55 rooms, with one apartment to a floor. There are two wells for condenser water, two washers, and two 18 hp vertical refrigeration compressor sets. Fig. 2 is a view of one unit, showing the washer and compressors.

The washers are operated to maintain a predetermined dew point, and preheater and reheater coils are automatically controlled. Although there is a separate radiator heating system and a central boiler plant, the reheaters in the air conditioning ducts (steam supplied from same boiler) can be used for supplying warmed air in cold weather. Control of the amount of cooled air to the rooms is left to the tenant of each apartment.

Operating cost figures are now available on this building. On hot days the equipment is operated between 6 and 7 hours and the total power requirement for both units is 40 kw-hr. With a special power rate of 1½ cents per kw-hr, the cost of operation is very moderate.

In another apartment house, a basement unit type of installation provides conditioned air for the apartment used by the owner of the building. This is a 6 ton job and supplies conditioned air to various rooms on two floors. In this case a recirculation system is used, since the space conditioned is occupied by one family only.

A simple room unit installation with central compressor located in closet space has been installed to

*Engineer, New York, N. Y.

[1]South America's summer occurs during North America's winter, of course.

condition the rooms on the top floor of a 6 story apartment building.

Three Types of Systems Used

It is interesting to observe from these representative jobs that three general types of air conditioning have been employed in solving apartment air conditioning problems:—(1) the large central system, using washers, preheaters, reheaters, duct distribution, etc.; (2) the central unit with direct expansion coils and duct distribution; and (3) the individual unit arrangement where a separate compressor serves a group of room conditioners. A fourth general type, the completely self-contained unit with compressor and fan, has not been widely used here for apartment house conditioning. One important reason for this is that the self-contained unit finds greatest utility in existing buildings rather than in installations in buildings under construction. Argentine has been enjoying a building boom of major proportions, and since this market is ready at hand little attention has been given to the possibilities in the older buildings. As has been learned in the United States, it takes a devastating building depression really to demonstrate the potentialities in selling equipment for existing buildings.

Reasons for Air Conditioning Apartments

Some architects hold the opinion that it is not essential to condition apartment houses, since most people who can afford to live in a conditioned building spend the summer in the country. However, the summer here is 7 to 8 months long, many people of means are kept in the city by business, and with a conditioned apartment, some families will spend a greater part of the hot season in the city than otherwise. Perhaps the best answer to the question of whether this splendid start in apartment house air conditioning is likely to continue in greater volume is that every conditioned apartment visited was completely rented, and those apartments under construction had signed many leases in advance.

Sprinkler Heads Smother Range Hood Duct Fires

By **H. G. Schaefer**

THE ventilation of a large kitchen is built around the range hood exhaust system. The greatest strain put upon the system is the periodic burning of the congealed grease which condenses on the duct surfaces. To withstand this abuse, it is generally conceded that heavy black steel plate be used in the construction of the ductwork; often, where physical conditions dictate the requirement, the duct is insulated with magnesia blocks to protect surrounding construction from the heat of a fire within the duct.

Such duct fires are generally intense. Since most kitchens use live steam for the operation of the cooking equipment, it is customary to install a live steam jet, controlled by a valve conveniently located, so that the fire may be smothered. Some one is generally sufficiently interested to see to it that the valve is painted a bright red to distinguish it as a part of the fire protection system.

The thought given to this jet equipment sometimes causes the designing engineer to go into a rhapsody in specifying an elaborate system of piping "properly supported and graded", within the duct, with so-and-so-many holes, so-many-sixteenths in diameter, not more than so-and-so-many inches apart, all pointing to the geometrical center of the duct section, to assure an even and homogeneous distribution of the smothering steam.

The trouble with all this careful forethought is that it comes under the classification of "Love's Labor Lost". There is no need to expound, to the heating and ventilating fraternity, on the subject of temperament; this ingredient in the synthesis which results in the Human Being, is the bane of the lives of us all. And it reaches a crescendo when incorporated in a chef.

The term "chaos" was coined by Noah Webster himself in a kitchen during a duct fire. The emotional outbursts, the mustachios waving in the breeze, the guttural cries—all to the obbligato of the roar of the burning grease—achieve an effect calculated to bring the chagrin of futility to an impresario of grand opera. And all this time, of course, no one has thought of the steam smothering line, so thoughtfully provided. The brilliant red paint of the control valve itself has turned a livid gray, in sheer mortification.

And so, we come to the point. Several years ago, some thoughtful chap sat up nights developing a most useful device known as a sprinkler head. It is a simple matter to run a line of pipe into a duct, with tees left at strategic points, and to make up, into these outlets, such heads set to fuse at a slightly higher temperature than they are normally provided for. The famous red valve is sealed open with a milk can seal, and the installation is complete, and waiting day and night, with never even the implication of human frailty, to choke off a fire as soon as it starts.

While no one can be depended on to open the smothering valve, you may be sure that the open valve will be shut off as promptly as possible, so that the boys can hear themselves tell each other all about what happened.

One of 1934's Biggest Jobs—

Rebuilding Chicago Stock Yards

By **Walter A. Becker***

One of the largest 1934 construction projects was the rebuilding of the Chicago stock yards after the fire last May. Because of the live stock exposition held every December, it was necessary to push the job through in record time . . . This article describes the heating, ventilating and air conditioning of the industrial, commercial, hotel, restaurant and exhibit buildings, and the International Amphitheater. The heating and ventilating exposition is to be held in the Amphitheater next January, it has been announced.

ON MAY 19, 1934, a cigarette thoughtlessly flipped into the cattle pens of the Chicago union stock yard caused property damage estimated at more than $3,000,000, the destruction of a dozen famous structures and numerous minor buildings.

The program of reconstruction involved the rebuilding of industrial, commercial, office, hotel, restaurant, and exhibit buildings, was scheduled for completion by last December 1 because of the International Live Stock Exposition held in the Amphitheater from December 1 to 8, was completed ahead of schedule. Mr. A. Epstein, structural engineer, designed and supervised the rebuilding.

The heating and ventilating equipment installed in the various buildings was given careful consideration to assure economical and efficient operation. As before, all buildings are heated with low pressure steam and have vacuum return main systems to handle the condensation. Steam is supplied from a bleeder turbine in the central steam plant and all condensation is handled by the main vacuum pumps situated there. Additional vacuum pumps at stations on the premises are cut in as booster pumps only when needed.

The steam and vacuum service mains are run in an extensive system of pipe tunnels, the source of steam supply being more than a mile from the service entrance to some of the buildings. Steam service mains in pipe tunnels are covered with 85 per cent magnesia pipe covering, and are dripped at close intervals through combination float and thermostatic traps to assure dry steam at each building. Expansion in the lines is taken up by slip expansion joints.

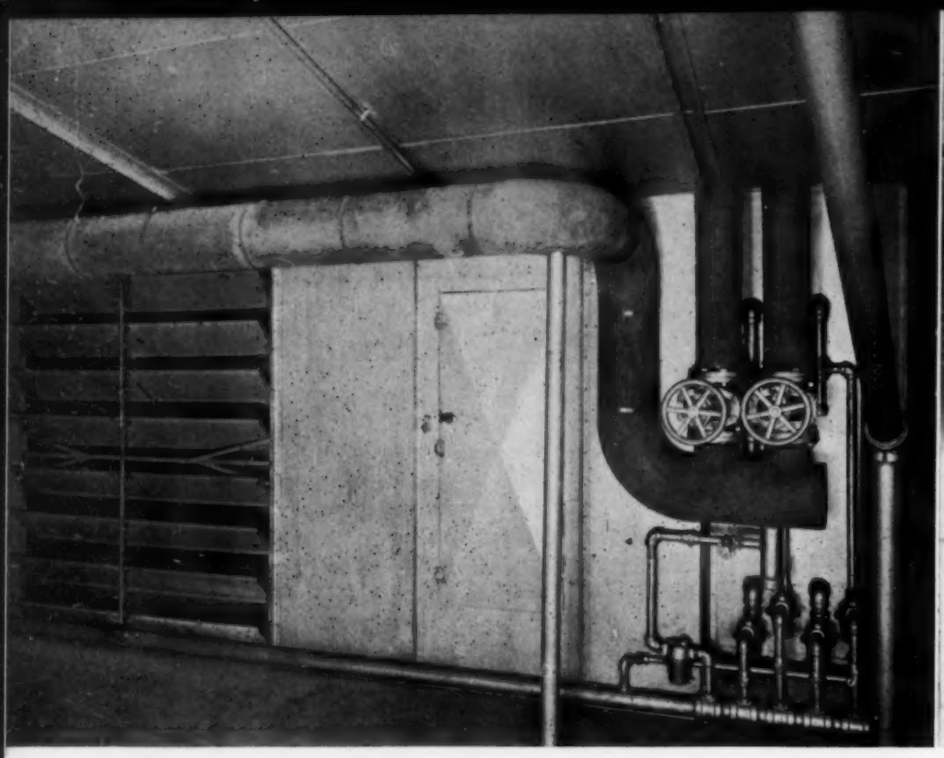
Heating the 7200-Seat Amphitheater

The buildings that offered the greatest difficulty for mechanical design were the International Amphitheater group, which comprises three sections—the Amphitheater (seating capacity 7200), and two exhibition buildings. The exhibition buildings are two story concrete structures 165 ft by 310 ft and adjoin the Amphitheater on each side. After carefully considering the type of heating equipment to be installed in the group, and as mechanical ventilation was required for the Amphitheater section, it followed that it was only practical to utilize this air for heating as well as ventilating that space; therefore an adequate blast system was installed. Due to the area of the exhibit halls and the use the name implies, unit heaters were decided upon as being the most desirable equipment to use for efficiently heating these spaces.

To ventilate the Amphitheater it was necessary to install fans and heating coils for a capacity of 180,000 cfm of air. As the size of the equipment required to handle this amount of air would attain such large dimensions and a correspondingly large area for erection, several tentative sketches were worked up in order to ascertain where it would be best to locate the fan rooms to conform to the architectural design. These sketches indicated that from an installation and operating viewpoint, the best plan was to divide this quantity of air and install four independent systems, each to handle one-quarter of the building. This simplified the design in general and allowed for symmetrical location of a fan room in each of the four corners of the building above the stair wells at the gallery level. This location for the fan rooms avoided the use of valuable exhibit space on the lower floors and did not interfere with the seating and passage arrangement at the main floor and gallery levels of the Amphitheater.

With the fan rooms so located it became an easy matter to design a separate system of supply ducts to handle each quarter of the building. The general scheme was to install four main trunk ducts overhead, two ducts extending halfway along each side. From these ducts vertical flues drop parallel with the arch columns and have outlets installed at the gallery and main floor levels. Two main trunk ducts were also installed along each end of the building, and located at the soffit of the gallery, each duct extending halfway across an end. Vertical flues run up to the gallery level and down to the boxes. This method of distribution eliminated unnecessary and expensive sheet metal work and affords uniform air

* Engineer formerly on staff of A. Epstein, structural engineer. Now associated with Grinnell Co., Chicago, Ill.



circulation throughout the Amphitheater. Outside air is taken in from the roof through intakes installed in the rear wall of each fan room, which walls are not visible from the ground surrounding the building, obviously desirable from an architectural standpoint. Recirculation was provided for use when the Amphitheater is unoccupied and it is desirable to keep it tempered only.

The supply fans and the heating coils are erected in the fan rooms for a "pull through" arrangement; that is, the heating coils are at the inlet side of the fan. A plenum space is provided at the cold air side of the heating coils and the inlets to it are arranged so that outside air, recirculated air, or a mixture of both, may be handled. The outside air intake is equipped with a wire mesh screen backed by a set of stationary weather louvres. A set of louvre dampers is installed in back of these. A second set of louvre dampers is installed in the recirculating opening in the plenum space housing. Both sets of dampers are for manual operation only; they can be set for the operating conditions previously mentioned although the intended purpose is to allow for total recirculation for the "heating up" period only. The return air is taken back through screened recirculating openings provided in the partition which divides the fan rooms from the Amphitheater (see sketch).

A 179 rpm fan having a capacity of 45,000 cfm of air against 1¼ in. static pressure is installed in each fan room. Each fan is full housed, single inlet, single width and is arranged for clockwise rotation with top horizontal discharge. The wheel is of the forward curved multiblade type; there are two bearings (one on each side of the fan) and an overhung pulley for V-belt drive. Each fan is driven by a 20 hp, 1,200 rpm variable speed motor equipped with slide rails for belt adjustment. An automatic starter and a speed controller with low voltage and overload protection are installed in each fan room to operate the fan over a range of speeds varying from full speed down to a 50 per cent reduction.

The heating coils (totalling 1,512 sq ft of heating surface in each fan room) are made up of cast iron heating surface arranged two stacks deep and two tiers high, erected on 18 in. high concrete piers located in the plenum space. Each tier of coils is tapped at both ends for top supply and bottom return. Supply and return headers are installed in each fan room, branch connections being taken from these headers and connected to the respective tapping at each end of the heating coil. A combination float and thermostatic trap of ample capacity to handle the condensation is installed in the return connections at both ends of the heater. Each tier is center vented as well through thermostatic traps, as shown in one of the sketches.

As mentioned, two main supply ducts lead from each fan room, one extending half the distance along a side of the building and the other half of the distance across an end. The ducts along the sides are installed above the haunch of the main steel arches and are supported on 2 in. angles spaced approximately 4 ft 6 in. on center which span two roof purlins at the eave. Erection of the ducts in this location necessitated raising the roof at the eaves. Nine vertical flues drop alongside of the steel arch columns from each main supply duct and have an outlet at the gallery and at the main floor level (see sketch). The ducts extending across the ends are dropped through the fan room floor and are run at the soffit of the high point of the gallery. Five vertical flues rise out of each duct to a high point which is 7 ft above the top row of gallery seats to the bottom of the outlet. The pitch of the roof varies this height on the east and consequently the height of the flues is lowered so that the top level is in keeping with the rake of the roof. Five flues are also taken out of the side of each duct and parallel the soffit of the gallery down to the box space outlets.

This system of distribution provides 114 supply openings in the Amphitheater, 56 at the gallery level and 58 at the main floor level and in the boxes. At the face of each supply opening is a ½ in. wire mesh screen backed by a set of adjustable horizontal louvres. The distribution of air is based on the seating arrangement to obtain a practical and positive

LEFT TO RIGHT

Steam and return connections for the heating coils in one of the four fan rooms in the International Amphitheater. One of the two supply ducts leading from this fan room is seen at the top of the picture, and at the left of the view is the return air intake

The large duct leading from the foreground supplies half the end of the Amphitheater and the duct leading from the background supplies the other half from another fan room. The flues taken from the sides of these ducts lead to the box space outlets and flues taken from the tops of these ducts supply the gallery seats. The small duct is the toilet exhaust

One of the two main supply ducts from one of the four fan rooms installed above the haunch of the arches and supplying one-half the side of the Amphitheater. Flues dropping down from this duct have outlets at the gallery and main floor levels (see sketch)

circulation where most needed. Over 60 tons of copper molybdenum iron sheets were used for the duct work.

Exhaust ventilation is provided by propeller fans in the monitor directly over the center of the arena. There are two exhaust systems for the toilets, one for the west end and the other for the east end. The horizontal ducts for the toilet systems parallel the Amphitheater supply ducts at the soffit of the gallery at each end. The exhaust fans are on platforms suspended from a toilet room ceiling and the discharge ducts run to the two south fan rooms where they extend up through the roof and are capped with gooseneck bends. Each fan has a capacity of 2100 cfm, is direct coupled to a ¾ hp, 1800 rpm constant speed motor, and has a multiblade forward curve wheel.

Units Heat Exhibition Buildings

Suspended unit heaters in both stories of each exhibition building are situated close to the exposed walls and spaced at approximate 50 ft centers; there are nine units in the first story and twelve in the second story of each exhibition building. The combined air capacity of the units in each space is sufficient to provide four air changes per hour.

A pipe tunnel along the entire perimeter of the building accommodates the steam and return piping. Steam and vacuum service mains enter the tunnel at the northwest corner of the north exhibit building. The steam service main is continued through the entire length of the west pipe tunnel and is used as a steam header. Four steam mains are taken off the service main in the pipe tunnel of the Amphitheater group, one for the north exhibit building, one for northwest and northeast fan rooms, one for southwest and southeast fan rooms, and one for heating system in south exhibit building. Each steam main is valved separately so that each exhibit building is independent of the blast systems in the Amphitheater.

The steam mains for the exhibit buildings are installed in pipe tunnel. Branch mains are taken off these mains to supply the few direct radiators in the corridors and small rooms of the Amphitheater. This arrangement enabled the isolation of the fan blast heaters from the general heating system. The steam main for the blast heaters in the northwest and northeast fan rooms rises out of the pipe tunnel and continues at the first story ceiling of the north exhibit building adjacent to the arch columns. A main riser is taken off for each fan room and is run up parallel to an arch column. An identical arrangement is used in the south exhibit building for the southwest and southeast fan rooms. Return mains from the fan rooms parallel the steam mains to pipe tunnel where they are connected into the common vacuum return for this building group.

Expansion in the steam and return mains in the pipe tunnel is taken care of by means of double traverse slip expansion joints with anchor plates. Two joints are installed in service main which is 530 ft in length. A 50 in. radius pipe bend is welded into each overhead steam main at the center of the horizontal run. Steam mains are dripped at points of rise and in front of each expansion joint. Service main is also dripped midway between the two expansion joints. Each main riser is dripped at base. Drip legs half the diameter of the pipe being dripped but not less than 2 in. in diameter are welded in the line of each drip point. Combination float and thermostatic traps are used at drip points.

Piping for steam and return mains 5 in. and larger was welded, welding fittings being used. Piping 4 in. and smaller was screwed. Steam mains throughout the building are covered with four ply 1 in. air cell pipe covering; all steam risers are covered with two ply ½ in. air cell pipe covering.

Other Buildings Damaged

Although the famous Stock Yard Inn and the Exchange Building were badly damaged by the fire, it was unnecessary to replace all of the piping. The steam mains in the Inn are run in the attic space and were not affected by the fire or the heat, although the steam and return risers, being exposed, were affected in some cases and about 15 per cent had to

be replaced. Numerous radiator sections were badly cracked and 75 per cent of the nipples had loosened. This condition was identical in all buildings where cast iron radiators were used.

The steam mains in the Exchange Building are also in the attic space, but a water test proved that these had all loosened at the joints and had to be tightened. The condition of the exposed risers and radiators was practically as found in the Inn. Wall type radiators are used throughout the Exchange Building and only 90 sections out of 1,100 had to be replaced. The intense heat played havoc with the radiator valves and traps. None of these specialties were in condition to be re-used and all were replaced.

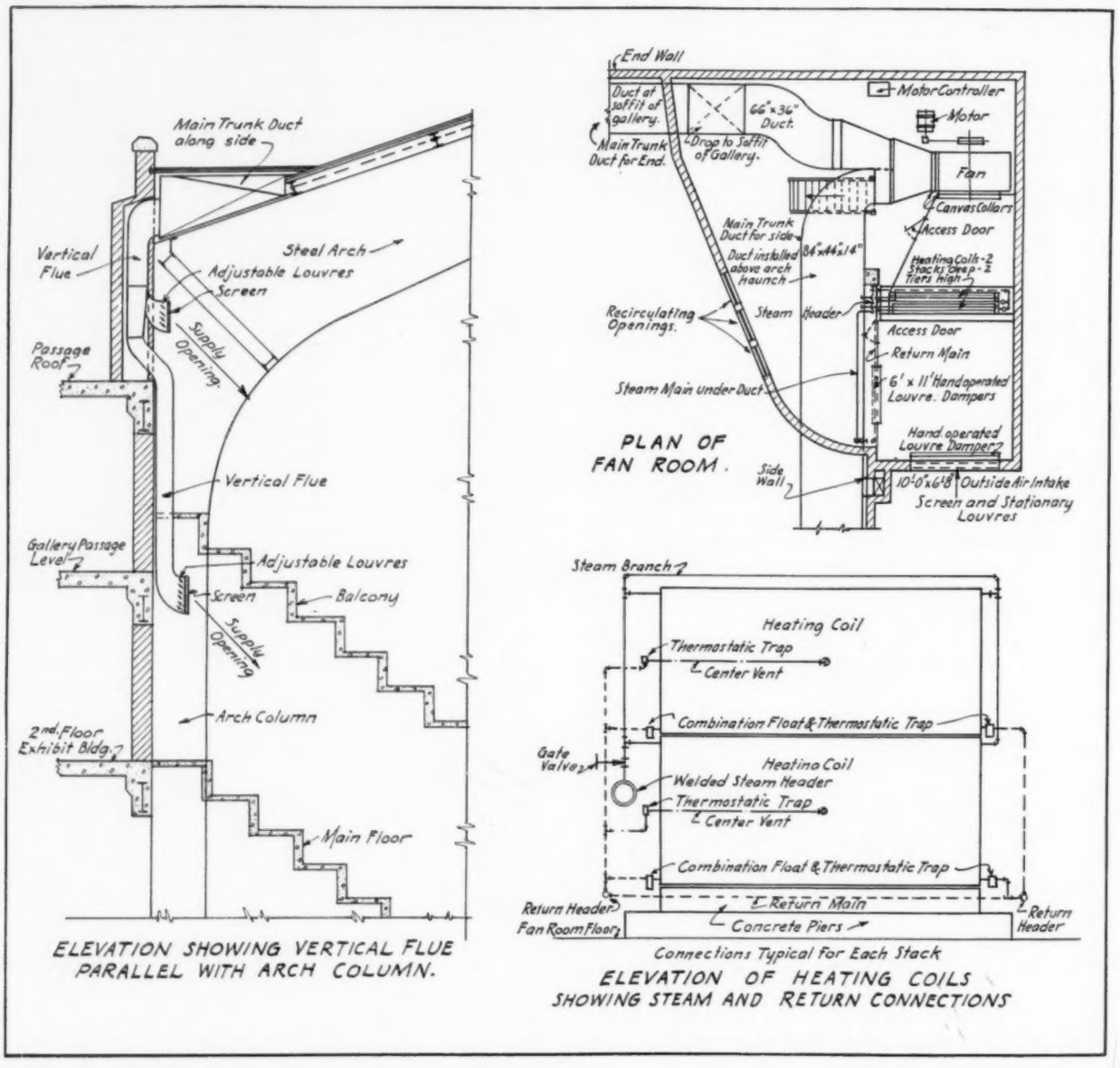
The condition of the heating systems in the Live Stock National Bank and the Record Building in which is housed the Saddle and Sirloin Club, was similar to that in the Inn.

As these four buildings were restored as they originally stood before the fire, the general arrangement of the heating system was not disturbed. Wherever possible radiators were rebuilt on the job and re-used. New radiators and piping were installed as needed, and all steam specialties replaced.

Restaurant Is Cooled

The Exchange restaurant building is constructed "air tight" with all window sash stationary. The dining, tap and lounge rooms are air conditioned, the 50 ton freon compressor, cooling coils and multiblade fan being installed in an adjacent basement room. The cooling coils and heating coils are erected at the inlet side of the fan, making this a "pull-through" arrangement. Three separate concrete tunnels under the floor handle the outside air, return air and the conditioned air, a vertical sheet metal flue running from the outside air tunnel to the intake on the roof. Return air intakes in the dining and lounge rooms connect to the return air tunnel. Two vertical sheet metal flues from the conditioned air tunnel to the roof are connected to two cork insulated (1 in.

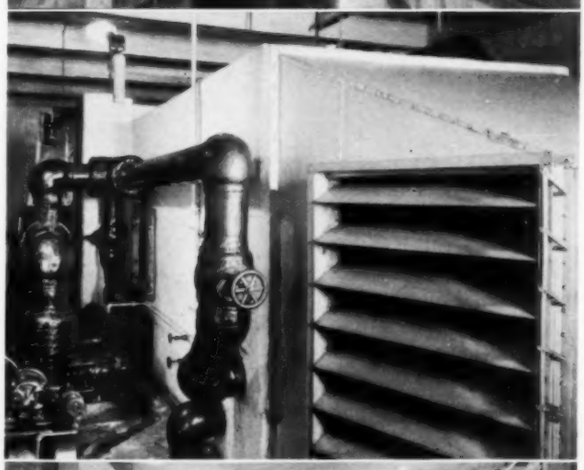
[Continued on page 195]

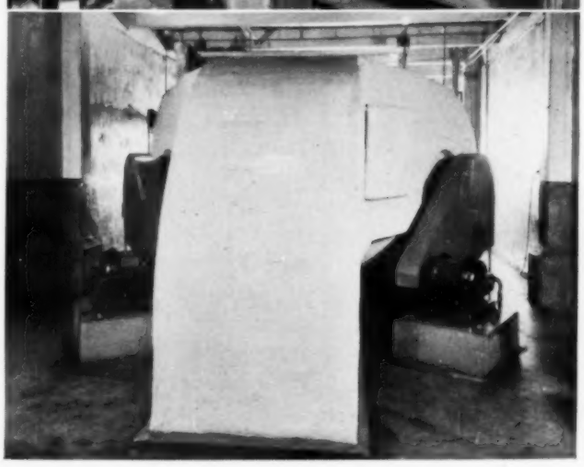


By **A. B. Wason***

Air Conditioning Aids in Manufacture of Ale

PRIOR to prohibition days, the brewer recognized the weather as a tremendous source of grief but comparatively little was known about overcoming the expensive and uncertain difficulties it caused. The rapid development of air conditioning during the last fifteen years has made possible continuous production, uniform quality, improved working conditions and big savings in the manufacturing process in many industries. Control of three of the essential elements of air conditioning—temperature, humidity and air distribution—have developed many industrial processes where the importance of air conditioning is widely recognized.





As in these other industries uncertain weather conditions no longer need be a problem in the manufacture of beer; with the proper equipment it is possible to produce ideal temperature, humidity and distribution of air to maintain constant air conditions throughout the year.

A case in point is the Dawes Brewery, Montreal, Canada, which was modernized by installation of a cooling and air conditioning central station system for the fermentation room used in the manufacture of ale. This room, approximately 180 ft long by 90 ft wide and 22 ft high, contains twenty large wooden vats with ale in different stages of process of manufacture. The ale in these open vats gives off a considerable amount of carbon dioxide gas and some moisture during fermentation. The rate of fermentation and evaporation varies, depending upon the surrounding air temperature and relative humidity.

Uniform control of room temperature, relative humidity and air circulation allows fermentation to take place at a constant rate and prevents evaporation. The treated or conditioned air enters the fermenting room below the intermediate floor and flows upward around the vats preventing accumulation of carbon dioxide gas and contamination of the air. The fresh, filtered, dustless air, free from spores and bacteria, supplied to this fermenting room provides proper ventilation, removes excess accumulation of carbon dioxide gas during the process of fermentation, maintains clean, healthy, working conditions.

The fermenting room requires a cool even temperature; unless the relative humidity is under control, excessive precipitation or condensation forms on the walls resulting in wetness of the floor, dampness and slipperiness generally.

There are always a few vats empty during the normal process of manufacture; by maintaining controlled temperature, relative humidity and air distribution the possibility of these wooden vats shrinking and consequently causing leaks is obviated.

Air conditioning of the fermenting room controls the rate of fermentation; thus the fermentation time for the manufacture of ale is uniform, the quality of the product is controlled with a resultant saving on investment of material in process. Steady production through the plant makes for economy in manufacturing costs, as uncertain production methods are eliminated.

*The Cooling and Air Conditioning Corporation, Division of B. F. Sturtevant Company, New York City.

Top to Bottom—Air conditioned fermentation room of the Dawes brewery. . . . The water circulating piping around the dehumidifier. . . . End view of the two fans, which are installed in duplicate to insure continuous operation

Installing Piping in Cramped Quarters

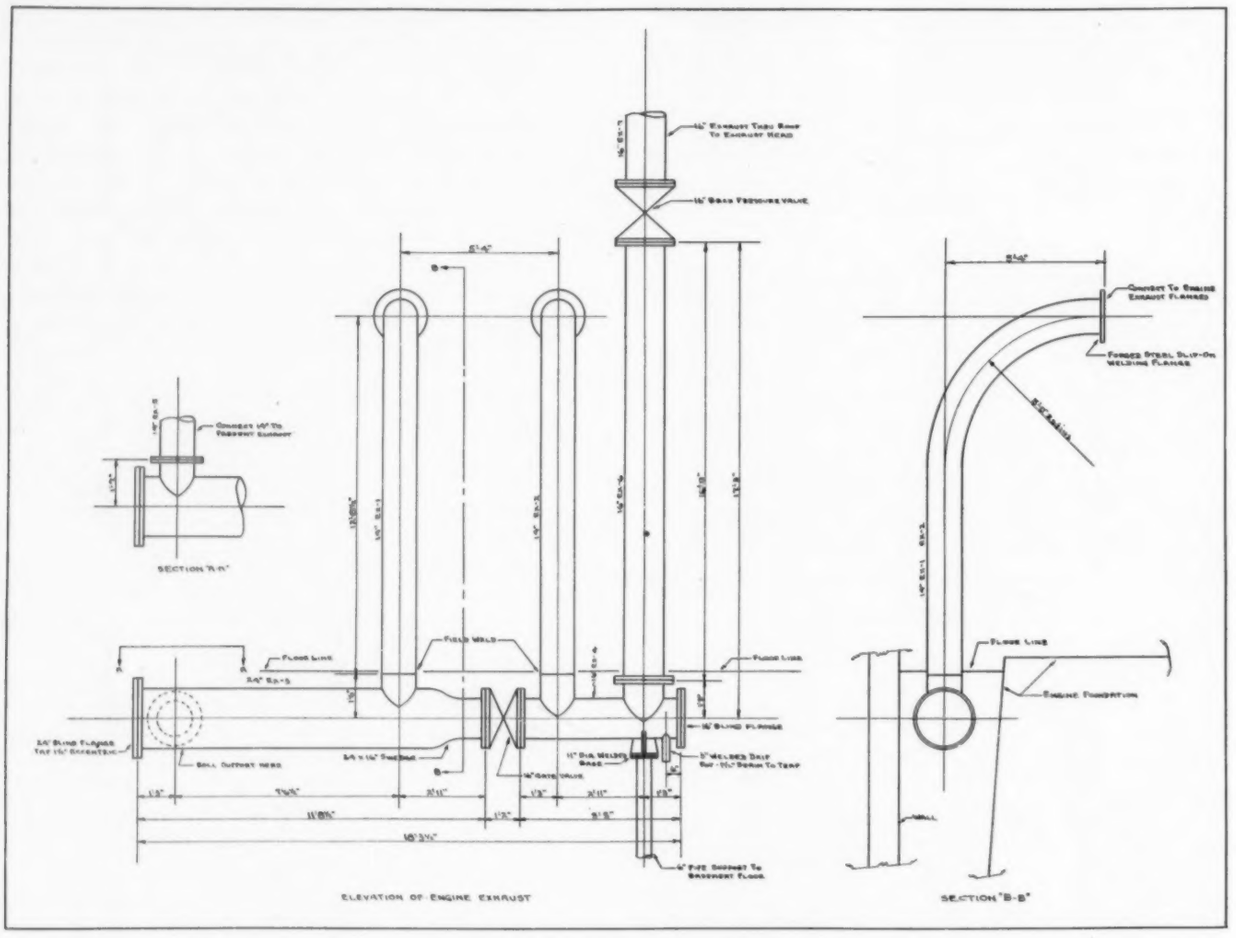
By **G. W. Hauck***

Fabricated Job with Flanged, Welded Joints Solves Problem

INSTALLATION of a new engine in a plant with limited available space presented a two-fold problem in locating and designing the exhaust header with its connections, and also in determining the type of joint best suited for both the conditions and economical final erected cost. This case indicates the advisability of study to determine the type of joint best suited for any piping job, especially where unusual conditions obtain.

Section "B-B" of the sketch shows the limited space for the installation of the exhaust header, caused by the proximity of the engine foundation to the building wall. Because of this confined space, it was considered desirable to hold to a minimum the number of joints in the header to eliminate maintenance difficulty as it is obviously a task to pull up bolts in large flanged joints where a line is located in cramped quarters between two walls. The shop fabricated welded headers were designed with a gate valve for sectionalizing as shown in the elevation of the sketch. The two headers and 16 in. valve were bolted and lined up and then lowered through the temporarily opened section of floor to proper elevation and position, being supported at the ends and center as indicated.

*Engineer, Crane Co., Cincinnati, Ohio.



Many piping jobs require careful study to determine the proper type of joint to meet the conditions. These are the details of an exhaust header which had to be installed in cramped quarters

The conditions prescribed the use of both flanged and welded joints for not only the run of the headers, but the outlets as well. For ease of installation and maintenance, the major portion of the main header was welded as outlined. The ends of the 16 in. and 24 in. headers, however, were provided with flanges for future extensions. Had the ends of these headers been welded shut, future extensions would have been difficult because position welds in the space between the headers and adjacent walls and floor above would have been extremely difficult if not impossible, whereas it is possible to make up bolts in the flanged joints on the ends with a ratchet box wrench.

Welded joints were selected for connecting the two 14 in. bends "EX-1" and "EX-2" to the 24 in. header "EX-3" and 16 in. header "EX-4." Flanged joints in place of the field butt welded joints would have interfered with the continuity of the floor line as the flanges on the bends would have projected above the floor line. The two 14 in. beveled end nozzles were shop fabricated in the headers as the limited space precluded the satisfactory shaping and welding of the ends of the bends directly to the headers "EX-3" and "EX-4"; however it was possible to accomplish the simple butt welds shown. In erection, the flanged ends of the bends were bolted to the flanged engine exhaust openings, which supported the bends with but a small amount of additional rigging until welded to the nozzles on the headers.

A flanged joint was selected for connecting the 16 in. riser "EX-6" to the header "EX-4" because the flanged joint is entirely below the floor and a welded joint at this point would have been extremely difficult of accomplishment, for considerable rigging would have been required to support and align the piece "EX-6" at the top, where the back pressure valve is located, in order to weld at the point of connection to the header "EX-4". With the flanged joint at this point, it was a comparatively simple job to erect the piece "EX-6" into position and conclude the riser with the section of piping through the roof with exhaust head.

Air conditioning is not confined to any geographical section, according to K. C. Porter in a recent article,[1] one reason being that the internal heat load of a building makes outside conditions insignificant as far as the need for air conditioning is concerned. A bank installation is cited as proof—in the spring refrigeration is used on the second consecutive day of 53 F weather and in the fall discontinued on the second day of 53 F weather. Two department stores with air conditioned basements operated the refrigeration equipment last November even though the mean outside temperature was 45.8 F and the maximum 67 F.

As to income to the utility from air conditioning, a typical figure of $12 per horsepower per year may be used, although this varies widely; from one restaurant the income was $9.36 per air conditioning horsepower while in the case of another it was $43.

[1] "The Big Air Conditioning Market," *Electric Light and Power*, February, 1935.

Rebuilding Stock Yards—

[Continued from page 192]

thick) distributing ducts which run parallel to the parapet wall on each side of the building. Short vertical flues drop through the roof and extend just below the ceiling to allow for side outlet grilles to be installed on each flue.

In addition, exhaust ventilation systems are provided for these three rooms and the kitchen. Propeller fans are employed in the lounge and tap rooms, and a multiblade fan with sufficient capacity to include the kitchen as well as the dining room exhaust is installed in a penthouse above the kitchen.

Industrial Type Buildings Have Units

Each of the industrial type buildings are heated with unit heaters suspended from the roof purlins, close to the outside walls, and spaced at approximate 45 ft centers. In buildings of sawtooth roof construction, a few heaters are hung in the sawtooths to compensate for the greater heat loss of this construction. Cast iron legless wall hung radiators are used in office spaces. Steam mains are run overhead in these buildings with branches to unit heaters taken off the top of the mains. Branches to downfeed risers are taken off the bottom of the mains and risers are dripped at the base through thermostatic traps. The vacuum return mains are run in pipe trenches below the floor and are united into a common return connected to the main vacuum service line.

The commercial and office buildings are heated with cast iron radiators (low window type in front of bulk heads in the stores and legless wall hung type in the office spaces). Steam and return mains are run in pipe trenches below the floor; consequently the systems are upfeed. Pipe trenches are covered with concrete slabs of such size to make them easily removable. Adjustable swivel ring pipe hangers are used to suspend the mains from cross pieces in the trenches spaced on 8 ft centers. Walls of trenches are slotted at the top to accommodate these cross pieces.

Pipe Insulation and Underground Piping

Steam mains and steam risers in all the buildings are insulated with air cell sectional pipe covering: four ply, 1 in. thick for the mains and two ply, ½ in. for the steam risers and branches. Return piping and radiator run-outs are not insulated except where they run underground in tile.

Very little steam and return piping is run underground. Where necessary wrought iron pipe in split vitrified tile laid up with full sand bearing is used, and the piping is insulated with two ply, ½ in. air cell pipe covering.

Wherever possible domestic hot water is supplied to the buildings from the hot water generators and storage tanks situated at central points on the premises. However, a few of the buildings have independent generators and storage tanks. Live steam is piped to these through separate valved mains. These tanks and generators are cross-connected to gas water heaters for summer use.

Air Conditioning

A MODERN radio broadcasting studio is like a thermos bottle—tightly enclosed and heavily insulated, with the primary idea of excluding vibrations and noises from outside but with the result that heat can escape but slowly since sound-proofing is also heat insulation. With perhaps seventy-five people and several thousand watts of lighting in such a room, there is definite need for cooling whether the month be January or July.

Such a system is installed in the new studios of Westinghouse Station KDKA in the Grant building at Pittsburgh, Pa., formally opened recently. The studios and associated offices occupy the entire third floor of the building, which floor was necessarily extensively modified to meet the specialized needs of such a tenant; sketch shows the floor plan. The studios are grouped in the central position and are flanked by the offices. The air conditioning system is designed to supply only the studios and associated rooms and not the offices.

The Air Conditioning System

The air conditioning machinery room occupies one of the inside corner office spaces from which air is distributed through ducts to the various studios. Both supply and return duct systems, together with a maze of conduits, pipes, and special service lines, are concealed above a suspended ceiling over the corridors.

In describing a closed circuit it is hard to know where to begin, but the fan is as good as any other point. This fan handles 7200 cfm against a head of approximately 1¼ in. of water and is of liberal design to minimize noise. The blades are of the backward curved type and the discharge velocity is a little over 1000 fpm.

Above the discharge of the fan is the bank of reheat coils, and above them, individual ducts are run to each of the principal studios.

The duct system was carefully designed and installed to avoid generation and transmission of noise. Just before the air reaches a studio it passes through a bank of sound absorbing tubes. These are square tubes, each approximately 8″x8″x16′ long, formed of heavy wire screen wrapped outside with sound insulating material. Enough such tubes are banked together to permit the passage of the required volume of air at a velocity of 600 fpm. All supply ducts are securely wrapped with insulation to prevent condensation and further to insulate against the entrance of noise.

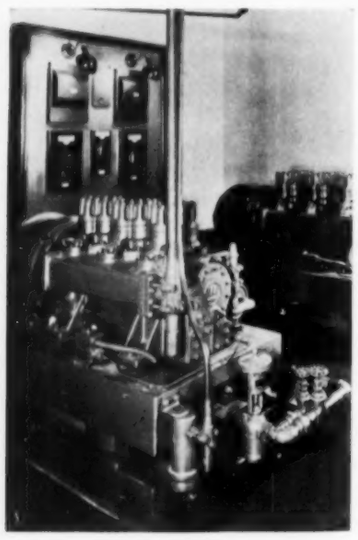
The air to the two largest studios enters through mushroom type outlets in the ceiling while in the others it enters through suitable grilles near the ceiling. Air is withdrawn near the floor and passes back to the machinery room through the return system and through another bank of sound absorbing tubes placed as near the studio as possible.

The return air, upon reaching the machinery room, first passes through the humidifier space. This consists of a large pan built into the duct and containing a steam coil. The pan may be filled with water when additional humidity is needed—an automatic steam valve is controlled by a humidistat in the fan casing to hold the water at a temperature which will provide the required relative humidity.

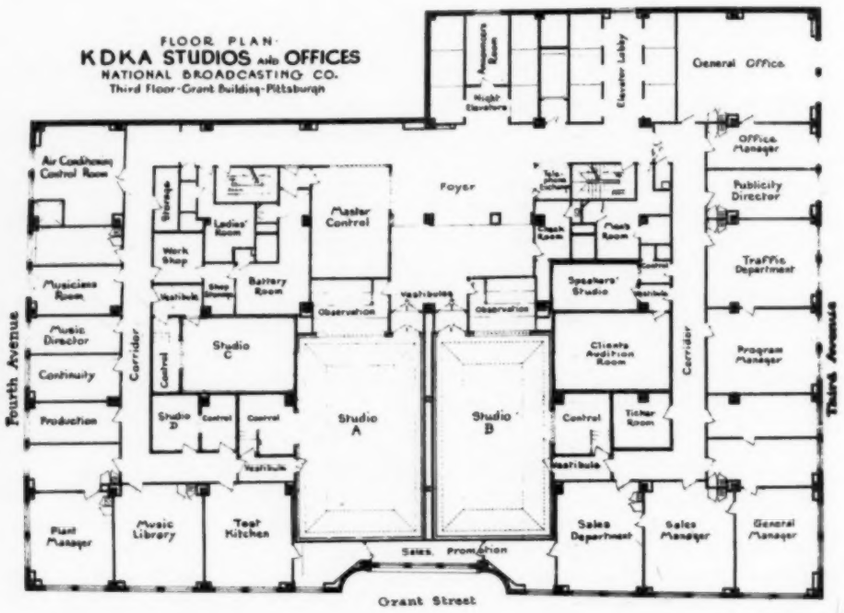
The return air system then joins the fresh air connection and is connected to the housing containing the cooling coils. There is a double bank of filters and then the cooling coils. There are four of them, two in series and two in multiple and after passing them the air goes to the fan for its next round trip.

Two 15-ton compressor condensing units supply the necessary refrigeration, one compressor supplying a pair of cooling coils.

*Manager, air conditioning department, The Danforth Company, Pittsburgh, Pa.



Two 15 ton compressor condensing units provide refrigeration for air conditioning



New KDKA Studios

By **H. W. Reding***

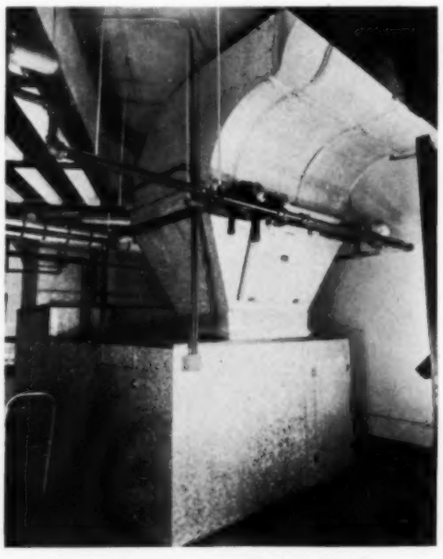
This outlines very briefly the physical layout of the system and will serve as a background for a discussion of some of the details of design.

Details of Design

Of the 7200 cfm circulated by the fan, one-third is drawn from outdoors, the remainder being recirculated. Since air must escape from the system, this is provided by discharging air from the smaller rooms into the corridor. The system was designed on assumed summer conditions outdoors of 90 F dry bulb and 73 F wet bulb. Studios are to be held under these conditions at 80 F and 50 per cent average relative humidity. To accomplish this the air leaves the cooling coils at a temperature of 57 and must enter the studio at a temperature not below 65 F even under maximum load conditions. As the load in any room decreases, the temperature of the air must rise to hold a constant room temperature. This reaches a maximum of something less than 80 F when there is no load at all.

Reheat coils supply heat as required and these are connected through motor operated valves which are under thermostatic control from the studio.

The thermostats are of the proportioning type. With outdoor temperatures of 90 F and above, the studios are held at a temperature of 10 F below that outside. As the outdoor temperature goes below 90 F the differential between indoor and outdoor temperatures decreases proportionately down to zero at 72 F. In other words, with an outdoor temperature of 90 F the studios are at 80 F. With 80 F outdoors the inside temperature is 76 F and with 72 F or less outside, the inside temperature is 72 F.



Another view of the air conditioning system for the studios

Since the big problem is one of cooling the year around, it is desirable to keep the temperature at the fan inlet low and allow the reheat coils to care for the temperature of air supplied to the studios. The tempering coil in the outside air supply is therefore bypassed by an automatic damper. The coil itself is the smallest which will not freeze in severe weather and it is to be turned on manually when the outdoor temperature reaches about 10 F. The damper operates from a thermostat in the fan casing to hold a temperature there of 57 F as far as may be done from full open to closed position of this damper.

The humidifier mentioned above is designed to hold a relative humidity of 40 per cent with a temperature of 72 F.

The system conforms to specifications prepared by the engineers of the National Broadcasting Company, which company operates the studio, and was designed and installed by the Danforth Company under the supervision of the N. B. C. engineering department.

Use of Backing Rings in Pipe Welding

SEVERAL readers have requested additional information about the backing rings used in the welding of the 600-lb, 850 F steam piping for the Connors Creek plant and described by Sabin Crocker on page 84 of the February 1935 H.P.&A.C.; the author supplements the information given in that article as follows:

The backing rings used were a ready-made variety in the form of a split band, somewhat tapered at the edges, and having a central ridge which helps in aligning the pipe ends and gives a nucleus for fusing in with the first bead. We tried making up continuous rings from strip rolled to shape in our own shop, after which the ends were welded together in the field where the rings were fitted into the end of the pipe with which it would be used. The practice at that time was to lay the closed ring aside after welding the ends together with the idea of using it later in the pipe to which it had been fitted. It was found, however, that closed rings of this kind would not fit so snugly as split rings having a reasonable amount of spring. Our present instructions to the field force for the use of such rings are:

Backing Rings: Backing rings approximately ⅛ in. thick by 1½ in. wide shall be used on all butt welds in pipe 2½ in. in diameter and larger, unless otherwise noted on piping drawing. All burrs shall be removed from the inside of the pipe ends before inserting the backing rings, so that these rings may be inserted with as little distortion as possible. Each ring shall be tack welded in place on one pipe end before the next length is brought into place. Split backing rings shall have their ends tack welded to the pipe and the joint of the ring shall be closed with a single, light bead. Split backing rings which fail to close by more than $\frac{1}{16}$ in. shall not be used. After attaching and closing the backing ring by welding, all spattered weld metal shall be removed from the inside of the pipe. Backing rings having a center rib that prevents movement in the pipe prior to welding, need not be tack welded to the inside of the pipe.

Cold Storage Room Piping Fabricated by Bronze Welding

FOR replacing the calcium chloride brine cooling coils in a cold storage room, it was proposed to make the coils of galvanized pipe welded at the joints with a low melting point bronze. Although this method of fabrication is not new, it has not been widely practised; therefore it was decided to investigate this type of joint thoroughly. Information was sought on the following questions:

1. Is it possible to find a good bronze with a melting point sufficiently low so that the galvanizing will not be burned off in the process of welding?
2. Will the dissimilar metals set up an electric couple which will cause corrosion?
3. Will the joint be mechanically strong?
4. Can the joint be made quickly and economically?

Although installations of bronze welded galvanized pipe are not common, one installation was located and inspected. The line had been in service about three years and had given no trouble. The chief engineer who was responsible for this installation was enthusiastic about this method of fabrication.

A plan of the proposed installation was submitted to one of the large manufacturers of welding equipment for comments and suggestions. At that time (1932) they reported that there were no authentic data on actual installations; however they did supply information to the effect that galvanized specimens had been bronze welded and tested for corrosion resistance and mechanical strength. They reported that salt spray tests had been conducted on samples of galvanized plate, some of which had been welded with steel rods and others with bronze rods. In every instance the bronze welded specimens showed less corrosion. The welding technique was then changed and additional specimens were made. A steel welded specimen and a bronze welded specimen were made for each set of conditions. One set of specimens was welded in an atmosphere of nitrogen; another set was made using molten fluxes; and a third set was made with a protective coating of aluminum paint on the galvanizing. All of these specimens were tested in a salt spray, but yielded results consistent with those made under normal welding technique. The electro-chemical couple set up by a bronze weld was not considered more potent than the iron-zinc couple created during steel welding. Zinc is electro-negative to both copper and iron and hence will be the metal that will corrode the more rapidly.

Tensile tests were run on three specimens; 1¼, 1½ and 2 in. I.P.S. Each specimen consisted of a short length of pipe with a threaded joint and a bronze welded joint. In no case did failure occur in the bronze welded joint.

So far, nothing had been found to condemn the proposed bronze welded construction, but it was decided to run corrosion tests to make doubly sure. One bronze welded specimen was placed on the roof for exposure to the weather. It was examined from time to time and at the end of 2½ years, it was removed, and sawed in two to allow an inspection of the inside surface of the pipe (see Fig. 1). Only a very small area of rust developed around the edge of the bronze weld where the galvanizing was altered.

Fig. 1—Test specimens

Another specimen was immersed for 25 days in a solution of calcium chloride brine at room temperature. A qualitative test of this solution indicated no copper or iron present, but that some zinc had entered solution. A visual examination showed no effect of the brine on the weld. In another test after 53 days of exposure to a solution of calcium chloride brine at room temperature, the solution was analyzed qualitatively. Iron and zinc were present in the solution. The bronze welded specimen was not visibly affected.

It was then decided to proceed with an actual installation. Double random lengths of hot galvanized steel pipe with scarfed ends were put in place on the coil supports, and as many joints as possible were made by rolling the pipe as the welding progressed. The terminal joints and the joints at the return bends were of necessity made with the pipe in its final position, the welder following the joint around the circumference of the pipe. All of the welding was done with an oxy-acetylene torch with the flame carefully regulated to confine the heat to as small an area as possible. A standard bronze welding rod having a melting point of 1625 F was used. A commercial flux was used in making all welds. When the coil was complete, it was tested with water at 85 lb pressure. The entire installation had 261 welded joints and only three small leaks were found which were easily repaired.

In welding galvanized pipe with either bronze or steel rods, the zinc is either vaporized or converted into zinc oxide. These minute metallic particles diffuse and contaminate the air. If much welding is in progress at one

By **T. F. Hooker**

Mechanical engineer, Eastman Kodak Co., Rochester, N. Y.

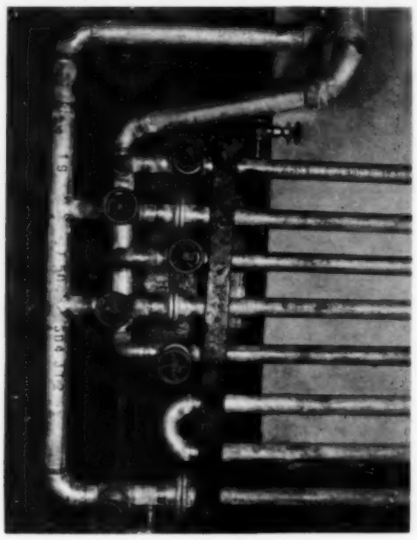


Fig. 2—Control end of the cooling coil with brine supply and return mains from overhead

time or if the ventilation is poor, additional ventilation or gas masks must be supplied to protect the welders from the toxic effects of the zinc in the air.

The bronze welded coils have been in continuous use since April 1933 and have given trouble-free service; frequent inspections show that the joints are practically corrosion-free. Late in 1933 another cold storage room was overhauled and new coils were installed with bronze welded joints, and early in 1934, a 4 in. line carrying cold water at 85 lb pressure was fabricated. No difficulty was experienced with the erection of any of these installations and although none of the systems have been in service long, it is felt that long life may be expected with a minimum amount of maintenance.

The advantages of a bronze welded galvanized pipe installation, as the author sees them, are:

1. The joint is resistant to normal corrosion.
2. The joint is physically strong.
3. There are few initial leaks. This is of special importance for lines carrying fluids which are hard to confine under pressure.
4. Cost of installation is low on pipe above 2½ in.
5. Resistance to fluid flow is low due to the use of streamline fittings.
6. Covering is simplified.

School Heating Costs, Other Data, Presented in Graphical Form

A STUDY made by Prof. R. A. Norman and undertaken for the purpose of obtaining data to enable comparison of the cost of heating and ventilating school buildings was recently issued by the Iowa engineering experiment station of Iowa State College, Ames, Ia.[1] The curves and explanatory material shown here are from the report of this investigation.

Twenty-four school buildings located in 13 different cities and towns were included in the survey. Collection of the data covered 1931-32 and 1932-33, but only part of the schools were included in both years.[2] Average temperatures for the different localities were obtained from the United States Weather Bureau at Des Moines. Systems of ventilation included the central fan, the unit, the window method, and combinations of those systems, while all used steam for heating. The types of heating systems included one pipe gravity, vacuum return line, and vapor with alternating return trap systems. Two schools used high pressure steam with steam engine driven fans, and four schools heated two or more buildings from one heating plant.

In plotting the curves the data were corrected or changed so that all the data plotted were based on the same or standard conditions, to permit comparison of one school with another. To illustrate, take the coordinate, "Tons of Coal per Season, Btu and Degree Day Basis," means that the coal has a heating value of 10,000 Btu per lb as received and the degree days for the season was 5700 (5400+6000÷2).

Fig. 1 shows the relation between the season coal consumption and the direct radiation. This curve may be used for schools having central fan or unit systems of ventilation. The window system of ventilation does not apply to this curve since the air for ventilation is warmed by direct radiators.

Fig. 2 shows the relation between the season coal consumption and the combined direct and equivalent direct radiation. If the amount of equivalent direct radiation is not easily obtained but the direct radiation is known, the equivalent direct may be obtained by referring to Fig. 8. In schools ventilated by the window method the total direct and equivalent direct radiation is the same as the direct radiation.

Fig. 3 shows the relation between the season coal consumption and the enrollment capacity.[3]

Fig. 4 shows the relation between the season coal consumption and the cubic contents of the building.

[1] "A Study of the Season Cost of Heating and Ventilating a Group of Iowa Public School Buildings," by R. A. Norman, Professor of Mechanical Engineering, Iowa State College, Ames, Iowa.

[2] The section of Iowa from which the data were taken gave an average of 5400 degree days for the season October 1, 1931 to May 15, 1932, and from the same period of 1932-1933, 6000 degree days.

[3] Enrollment capacity is used as a coordinate in the curves because it gives a more accurate comparison of different schools than the actual or maximum enrollment as furnished by the schools. Most schools gave enrollments less than their maximum capacity. Enrollment capacity is based on 225 cu ft per occupant in class rooms and study halls and 500 cu ft per occupant in laboratories. Auditoriums not used for study purposes are considered the same as study halls. All rooms are considered as having a standard ceiling height of 12 ft.

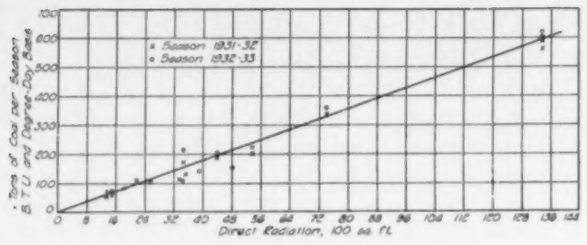


Fig. 1—Relation between tons of coal per season and direct radiation in square feet

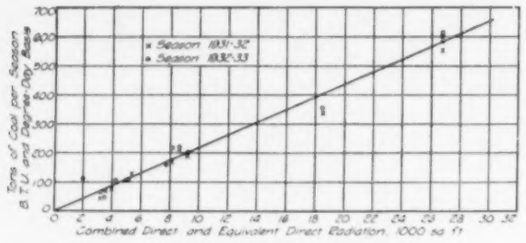

× Season 1931-32
• Season 1932-33
Tons of Coal per Season B.T.U. and Degree-Day Basis
Combined Direct and Equivalent Direct Radiation, 1000 sq. Ft.

Fig. 2—Relation between tons of coal per season and combined direct and equivalent direct radiation

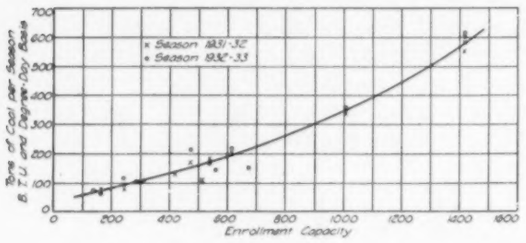


Fig. 3—Relation between the tons of coal per season and the enrollment capacity

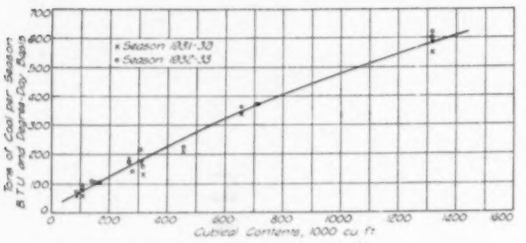


Fig. 4—Relation between tons of coal per season and the cubic contents of the building in cubic feet

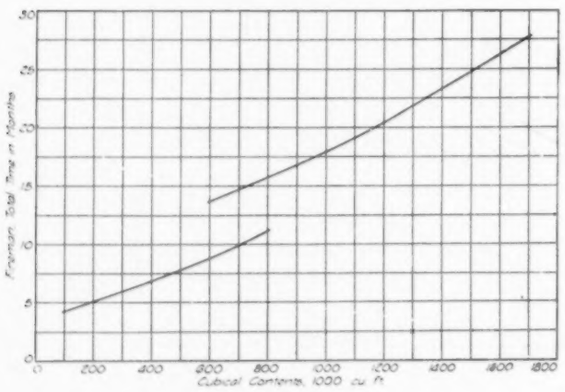


Fig. 5—Relation between the time of fireman in months and the cubical contents of the building

Fig. 5 shows the relation between the time of the fireman in months and the cubical contents of the building. In this curve the fireman's time is estimated from the information collected from the various schools. In the smaller schools the fireman was also the janitor and it was found in some cases that his duties were so many that the care and operation of the heating and ventilating system was neglected. In such cases the estimated time of the fireman was low. The curve in Fig. 5 is shown in two segments. The lower segment represents single unit buildings equipped with their own heating plant, in which case the fireman may have duties other than the care and the operation of the heating and ventilating system. The upper segment represents schools having two or more buildings heated from a single heating plant; those buildings that have high pressure steam heating plants with engine driven fans; buildings having large central fan systems equipped with washers; and those buildings large enough to require the full time of a custodian. It was assumed that only a part of the custodian's time was employed in the care and operation of the heating and ventilating system.

Fig. 6 shows the relation between the cost of power in kilowatt hours per season and the cubical contents of building. The information collected from the different schools regarding power cost was not complete nor accurate enough to use, since in most schools the power load was either combined with the lighting load or the power load included all the power equipment. The kilowatt hours actually consumed by the heating and ventilating equipment could not be separated.

The power consumption was calculated, based on a central fan system using a quantity of air for ventilation as determined from Fig. 9. Fig. 9 assumes that the amount of air required per person is 30 cfm. In calculating the load, the brake hp of the fan was taken from fan tables, adding 10 per cent for motor drive. The time of operating the fan was based on 7 hours per school day with two semesters of 16 weeks and a 5 day week. The lower curve gives the power consumption for a fan system operating as ventilating equipment. The upper curve is the same as above, except the power for operating the recirculating and spray pump and the increased load on the fan when air washer is used, is added.

In a unit system of ventilation the power consumption would be approximately 80 per cent of that shown on the lower curve, while window ventilation would require about 50 per cent.

Fig. 7 shows a comparison of maximum or total enrollment and capacity enrollment with the cubic contents of the building. The capacity enrollment and cubic contents are the same as used in the preceding curves while the maximum or total enrollment was furnished by the various schools.

These two curves indicate that for the smaller school buildings more of the total space is used for class rooms. This is especially true when the building is used for grade purposes only. The higher cubage takes in groups of buildings, or senior and junior high schools, where more space is given to laboratories, gymnasium, cafeterias, swimming pool, etc.

The other curves shown are self explanatory.

Results of the Study

The schools investigated were not chosen as representative schools of a given class or type but included grade, junior and senior high, and consolidated schools.

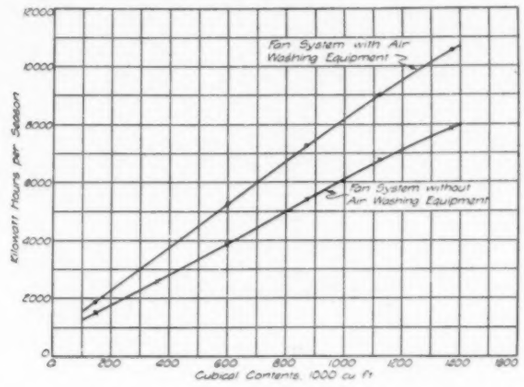

Fan System with Air Washing Equipment
Fan System without Air Washing Equipment
Kilowatt Hours per Season
Cubical Contents, 1000 cu ft

Fig. 6—Relation between power consumption in kilowatts per season and the cubical contents of the building

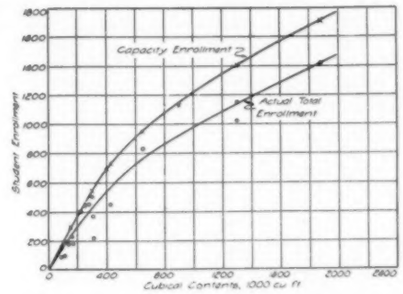


Fig. 7—Relation of the capacity enrollment and the actual total enrollment to the cubical contents of the building

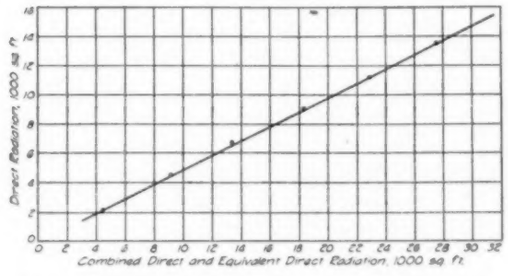


Fig. 8—Relation between the direct radiation and the combined direct and equivalent direct radiation

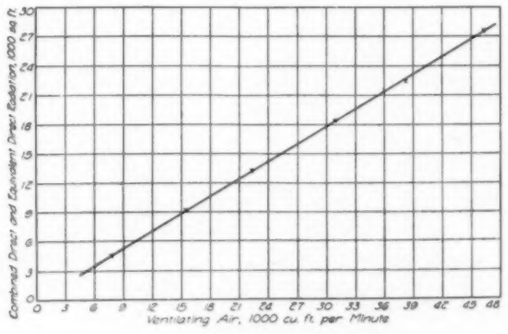


Fig. 9—Relation between combined direct and equivalent direct radiation and the ventilating air

Some of these buildings were of recent construction and their heating and ventilating equipment met the latest ideas of design and installation, while many of the buildings included in the survey were old and inadequately equipped from the standpoint of both heating and ventilation. *In comparing the fuel cost of the better equipped buildings with those where there was evidence that the heating was inadequate and with very little ventilation, it was found that the fuel consumption of these less favorably equipped buildings was in most cases as great as the better designed plants*—which means that it costs very little if any more for fuel to operate a well designed plant giving satisfactory heating and adequate ventilation than it does to operate a poorly designed plant of the same capacity furnishing unsatisfactory ventilation.

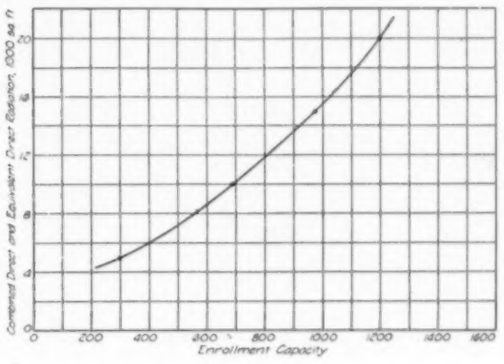


Fig. 10—Relation between the combined direct and equivalent direct radiation and the enrollment capacity

The cost of the service of the fireman is a considerable portion of the total cost of the heating and ventilating of a school building. Since the satisfactory operation of this equipment depends upon him, he should be qualified to perform these duties. This item of cost reaches its minimum ratio of total cost when the building is of such a size that it requires the full time of a fireman.

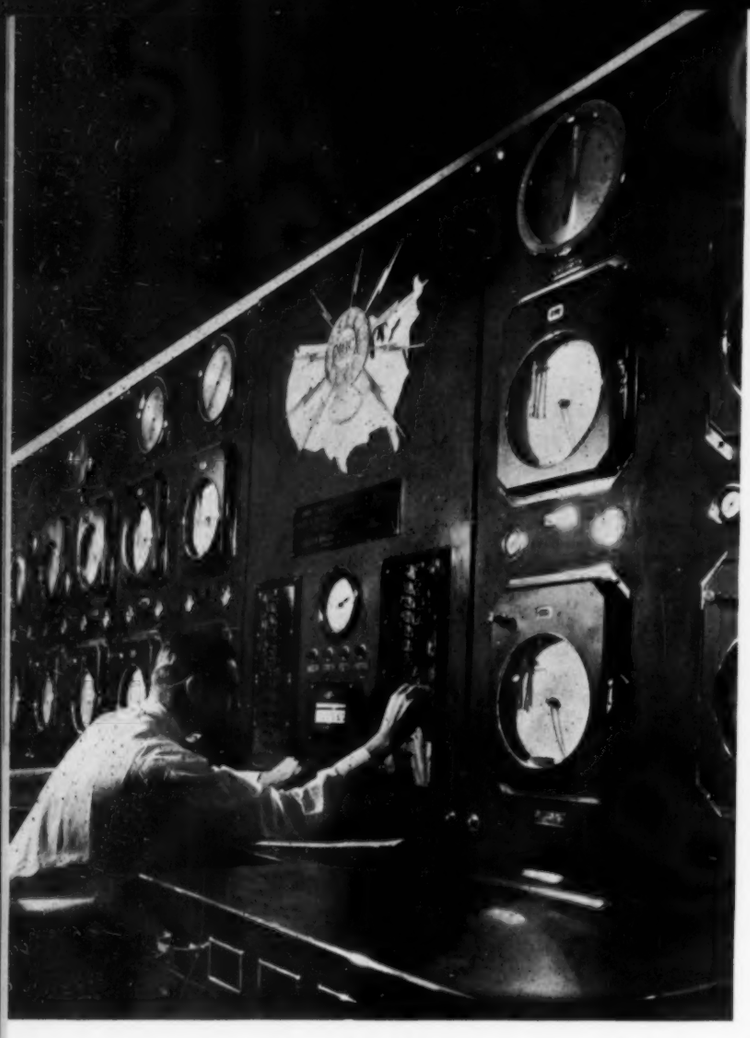
The cost of power runs proportional to the size of the building.

From the standpoint of heating and ventilating costs there is evidence that schools with a maximum enrollment of from 600 to 800 can be operated at least cost per student.

A 3 story building would offer less resistance to the flow of air than a 1 or 2 story building of the same capacity and therefore the power cost would be less for a central fan system of ventilation.

The cost of coal depends upon the grade of coal and the distance the school is from the mine. The selection of boiler and grate to burn cheap fuel should be investigated when new equipment is required.

Automatic temperature control installed throughout the building will save fuel by preventing overheating, which is one of the greatest losses in heating a building. Overheating and insufficient heating cause more illness in schools than any other item that can be mentioned in the problem of heating and ventilation.



Control panel for air conditioning system at N. B. C. studios, Radio City. The large dials keep a constant graphic record of air conditions in each section. The author is shown making a test

CHOOSING *the right* AIR CONDITIONING SYSTEM

Part 4—Fundamental Considerations

By **A. Warren Canney***

WHETHER central station or unit air conditioning systems, the same various combinations of essential equipment go to make up either. We will apply the term *unit system* to plants which do not utilize distributing ductwork, but the term "unit air conditioner" properly applies, whether ducts are to be used or not, to all equipment which incorporates two or more elements for effecting the functions of air conditioning. In differentiating we must be careful whether we say "equipment" or "system."

We need not now give further consideration to self-contained units with refrigeration as their use, as compared with the indecision in other matters, is usually obvious for one reason or another. Our subject here is the factors involved in the choice of a duct or ductless air conditioning system, each of which may have a central refrigeration plant.

Probably as good a place to start as any in the choice of a type of air conditioning system lies in determining whether ductwork is essential, merely desirable, or unnecessary, *i. e.*, whether units can be located within the space, because certain departures ensue once this decision can be made with respect to the character of the space. Something of a check list for establishing this point alone is as follows:

1. Appearance of units in space?
2. Can units be concealed?
3. Ceiling height with reference to air volume delivered per unit?
4. Connections for outdoor air feasible and of satisfactory appearance or concealment?
5. Are air conditioning requirements year-around or seasonal?
6. Is space available outside of room to be conditioned?
7. Do occupants face one direction?
8. Are they at rest?
9. What is the relation of heat sources or losses to possible methods of distribution with units?
10. What is the distribution velocity-envelope of the discharged air from units of required capacity?
11. Can sufficient units of sufficiently low air volume and discharge velocity be worked in satisfactorily?
12. Sound level increase if no ductwork?
13. If no ductwork, has infiltration been allowed for at the maximum capacity?
14. Does air distribution demand ductwork?

* Engineer in charge of air conditioning, Clyde R. Place, Consulting Engineers, New York, N. Y.

15. Does ductwork, insulation and furring counter-balance in itself the costs chargeable to additional units?
16. Will the system be required to heat or not?
17. Must the system supplement heating by delivering air at 70 F in winter?
18. Is exposed piping and draining permissible for exposed ceiling units?
19. If unit or units are concealed will proper access be had for maintenance?
20. If ceiling units in room are objectionable to appearance, is there floor space for floor units?
21. Is the ceiling flush?
22. What ceiling obstructions exist?
23. If a large volume of air from one or more units is indicated, will good distribution be obtained?
24. Can or must ductwork be concealed?
25. Will good distribution be obtained from the available space for ductwork?
26. Is regulation by the occupants permissible?
27. Is one large space with the same heat load characteristics, or several large spaces with the same heat load characteristics, or a number of spaces, small or large, with different heat load characteristics, to be served?

In evolving one or more designs for comparison, there are a considerable number of controlling factors. In existing buildings where rentable area sets a premium on the method, and where spaces are multi-partitioned, the indicated method can probably be immediately boiled down to either the circulation of chilled water from a basement refrigeration plant to air treating units at various floors or compressors on each floor for direct expansion units on that floor. Then, dependent on the extent of floor area to be served, the size of ducts as affecting appearance or reduced headroom must be compromised with the rentable area chargeable to the air handling apparatus.

If the partitioning is simple and there are no interior spaces, window units may be indicated. This can be done by replacing cast iron radiators with finned convectors in a cabinet which also contains the summer cold water coil. In this instance vertical chilled water and drain pipes are required at intervals along the outside building wall.

The matter of appearance is always important, and the degree to which appearance is to be sacrificed is a big stick over the air conditioning method. Mechanical and structural complications arise and if a clean-cut appearance is to be sustained, so simple an idea as replacing radiators with window units needs careful analysis. Lack of standardization of window and other building conditions has retarded development of standardized window units, although they are now available.

Relation of Air Conditioning to Heating System

The relation of the air conditioning system to heating should be established, if possible. The possibilities are:

1. Heating separate, as by direct radiation:
   (a) hand control of radiation
   (b) thermostatic control of radiation
   (c) by separate ventilating and heating unit or units
2. Heating to be included in and effected as a part of the air conditioning equipment
   (a) year round window units
   (b) by ventilating, heating, and cooling unit or units
   (c) reheaters at the central apparatus

At this point it is necessary to establish whether it is intended or considered ever desirable in the future to operate the air conditioning system in winter for humidification and ventilation, assuming the heating is to be independent. If so, it will be essential in winter for the system to deliver air at room temperature. It is very important either to know or decide this, as it has considerable to do with method. But before reaching any conclusion we should examine whether or not spring, fall, or winter air conditioning is either (a) desired, (b) not desired, or (c) of economic doubtfulness; for, if winter conditioning is not to be an accepted fact, and if summer is the first consideration, the approach to selection must be made from the summer angle.

This idea of "summer first and winter afterward" will perish in another quarter of a century. In the existing building it is now common and quite natural to approach air conditioning from the summer standpoint. The question then usually arises as to what increase in costs is involved for extending the services of the system into (a) spring and fall, and (b) winter.

There are two factors which help to settle this, although it will be frequently found that, first, a decidedly better investment balance obtains if spring and fall conditioning are incorporated and, second, the added operating costs for the use of this three-season equipment for winter also are usually small relative to the costs for ownership for the other three seasons.

These factors are, (1) whether the interior average heat load is large relative to the total capacity (total capacity being based on summer) during spring and fall, as well as to the total load during the intermediate seasons (the total load including the weather load); and, (2) whether design is being attacked from the standpoints of either (a) the best economic balance from the standpoint of advantages plus returns vs. initial outlay plus operating costs all told, as an investment; or (b) whether it is desirable or permissible to increased operating costs for the benefit of a lower purchase price; or, (c) whether first cost only is being considered for summer cooling with all other aspects of air conditioning thrown to the wind.

If summer air conditioning is to be the only consideration, and the equipment is to be entirely shut down except when the indoor temperature is excessive (no attention being paid to the indoor relative humidity or ventilation at other times) and also, if proper distribution can be obtainable, unit coolers may fill the bill, with no humidification or heating.

In place of providing all units with outdoor air connections, one or more units may recirculate air and one or more may have inlet air connections with a disc or other type of fan for direct exhausting, particularly if the heat load is high. The number of units connected outdoors will depend on the volume of outdoor air allotted as related to the total peak heat load.

This represents minimum air conditioning costs for the "evaporation side" for summer, the "condensing side" for refrigeration being an additional cost.

Another variation of this method consists of one large unit for one room with split cooling coils. If the load is very intermittent, as in some restaurants, it will pay to subdivide the refrigeration capacity with a number of machines, each cutting in on a section of coil with its own expansion valve.

If ductwork is not warranted, and if an analysis shows that but little cooling is required during spring and fall, then the matter of extending the useful period of the summer system at some additional cost can be dropped out of the picture.

Extension of Summer Systems

As a general thing, where the desire to acquire air conditioning originates from what might be called the "summer motive" an economic analysis should be made for buildings or spaces of Type 1 in the list below. Type 2 spaces need no analysis, as spring and fall operation is essential. Type 3 spaces may be analyzed in order to establish the difference between initial and operating costs between two methods as a matter of getting the best "financial fit" and Type 4 normally needs no such analysis, assuming the purchaser is interested only in summer cooling and dehumidifying.

Type 1	Type 2	Type 3	Type 4
Doubt	*Usually no doubt, must operate*	*Financial relation of initial to annual operating costs*	*Usually no doubt, analysis unwarranted if for summer only*
Cafeterias.	Theatres.	Office Buildings.	Residences.
Auditoria (occupancy intermittent).	Studios.	Banks, upper levels.	Apartments.
Ball Rooms.	Populated basements.	In general, interior spaces and those having high internal heat emission.	Hotel guest rooms.
	Department stores.		
	Restaurants.		

This list is based on the sole idea that the original intent was to acquire air conditioning for summer.

If the interior heat load during spring and summer is but moderate, then the supplementing of direct expansion coils by a recirculated spray for evaporative cooling may not be indicated. If the interior heat load runs high, or if heat losses to the outside are relatively low, then evaporative cooling probably will be warranted; but this in turn is dependent on whether the ratio of the latent to the total heat emission is high, for the higher this figure, the less becomes the value of evaporative cooling, which necessitates sprays and a recirculation pump. The problem is whether the additional cost for the apparatus with pump and sprays will be amortized by the savings accruing by the non-use of refrigeration during all those operating hours when the outdoor wet bulb is low enough for the minimum dewpoint required, all in correlation with the load characteristics. To go through this is a long process, involving weather bureau records. For repeated variable set-ups the author employs a series of pre-calculated tables for facilitating analysis.

The spray units are a little more noisy than the fan noise alone in direct expansion room units, hence the former finds wider industrial application and fewer "comfort" applications. Spray plus coils require more space, of course, and cost more, but direct expansion alone is not the economic solution for many comfort air conditioning applications. The relative quietness, the lower initial expense and the smaller space accounts for the wide adoption of the direct expansion unit without seasonal auxiliaries.

The third matter to consider seriously, after having determined the best economic balance between direct expansion alone vs. direct expansion supplemented by a recirculating spray or a recirculating spray alone is whether distributing ductwork is essential. A blunt affirmative answer would doubtless be correct for most applications, but there are many conditions which permit elimination of distributing ductwork.

Ductless units may be located within the room or adjacent to it so that the air discharge may be fixed when installed as either horizontal or vertical. The air leaves units in an almost square cross section which is not conducive to satisfactory distribution under all cases. Larger units, of course, have two or more outlets with guide vanes to provide a wider spread in the discharge velocity-envelope.

This practice is most likely to be successful where the ceiling height is high for the mass of air delivered and the room large enough to permit diffusion for the velocity of discharge. In general, the discharge of cool air vertically upward against the ceiling may provide excellent distribution, but this probability is oftimes seriously reduced by the necessity of locating units to the side of the room along a wall with a random velocity of impact against the ceiling due to varying unit heights and ceiling heights together with varying volumes and discharge velocities. Ceiling beams should be furred in.

Rooms which have "corners" or which have "spotty" conditions of heat liberation present difficulties to the ductless method.

Ductless methods which blow air horizontally on low ceiling heights usually must have recirculation to keep the air delivery temperature up.

Theoretical desirability of ductless distribution (from units directly) is seen by the fact that delivery temperatures 20 to 30 F below room temperature may be used, all, or nearly all outside air being used, because a simplified system results. But such distribution cannot be employed indiscriminately with units which rest on the floor at one side of a room; a flush ceiling seems essential, in any case. Such temperature differentials are out of the question for horizontal discharge except in rare circumstances.

Ductwork, then, is to be considered first from the standpoint of air distribution. Appearance of units within the rooms, space for units adjacent to the rooms, space for apparatus at some other point in the building, or the wisdom or economics of supplying several spaces from one conditioning apparatus, are entirely secondary and actually separate issues which are controlling within themselves and have nothing to do with the results vs. cost, or as to a unit system vs. a central station (duct) system. The real basis for the use or non-use of ductwork is air distribution.

Heating · Piping
and Air Conditioning

APRIL 1935

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Cooling Requirements of Single Rooms in a Modern Office Building

By **F. C. Houghten*** ***(MEMBER)***, **Carl Gutberlet**** **and Albert J. Wahl***** ***(NON-MEMBERS)***,
Pittsburgh, Pa.

THE growth of the practice of cooling single segregate offices in modern office buildings has resulted in a demand for better data for estimating the required cooling capacity. A study of this subject was made by the A. S. H. V. E. Research Laboratory in the Grant Building, Pittsburgh, during the summer of 1934, in cooperation with The Technical Advisory Committee on Heat Requirements of Buildings, D. S. Boyden, *Chairman*.

Three rooms on the 27th floor (2703, 2715 and 2718, hereinafter referred to as 3, 15 and 18), having northeastern, northwestern and southwestern exposures, respectively, were chosen for the study because of their variation in exposure and because they had been used in the earlier Laboratory study[1] of heating requirements. The investigation was directed toward the development of data of value in estimating the cooling capacity needed for individual rooms in office buildings, where the demands of the occupants are limited to the period from 8:30 or 9:00 a. m. to 4:30 or 5:00 p. m.

Test Arrangements and Procedure

The three rooms had approximately equal outside exposures and window areas. All inside partitions, including ceiling and floor, separated them from other unconditioned rooms. Table 1 gives characteristics of the rooms which may affect their rate of heat gain.

Table 1—Room Characteristics

ROOM 3 VOLUME — 1926.8 CU. FT.						
CONSTRUCTION	RIGHT WALL	LEFT WALL	CORRIDOR WALL	OUTSIDE WALL	CEILING FLOOR	WINDOW
	C	A	A	D	E	
AREA-SQ.FT.	197.0	187.3	81.6	58.3	197.8	40.8
AIR TO AIR *	70.5	74.9	32.6	18.2	76.1	46.1
SURFACE TO AIR‡	325.0	309.0	134.6	96.2	326.4	67.3
ROOM 15 VOLUME 1069.8 CU. FT.						
CONSTRUCTION	RIGHT WALL	LEFT WALL	CORRIDOR WALL	OUTSIDE WALL	CEILING FLOOR	WINDOW
	A	A	A	D	E	
AREA-SQ.FT.	115.2	110.7	93.0	55.1	109.6	37.9
AIR TO AIR *	46.1	44.3	37.2	17.2	42.2	42.8
SURFACE TO AIR‡	190.1	182.6	153.4	90.9	180.8	62.5
ROOM 18 VOLUME = 2005.8 CU.FT.						
CONSTRUCTION	RIGHT WALL	LEFT WALL	CORRIDOR WALL	OUTSIDE WALL	CEILING FLOOR	WINDOW
	A	A	B	D	E	
AREA-SQ.FT.	187.1	192.0	88.5	61.8	204.5	40.8
AIR TO AIR *	74.8	76.8	33.7	19.3	78.7	45.9
SURFACE TO AIR‡	308.7	316.8	146.0	102.0	337.4	67.0

* HEAT TRANSFER BTU PER HOUR PER DEG. FAHR. DIFF. AIR TO AIR — ENTIRE WALL
‡ HEAT TRANSFER BTU PER HOUR PER DEG. FAHR. DIFF. SUR. TO AIR — ENTIRE WALL
A-3" TILE, ½" PLASTER BOTH SIDES U= .400
B-4" TILE, ½" PLASTER BOTH SIDES, ¾" MARBLE & WAY UP ON CORRIDOR WALL AV U-.361
C-½" INSULATING BOARD ON 2X4 STUDDING U= .358
D-4" FACE BRICK, 4" COMMON BRICK, 4" TILE, ½" PLASTER, 15 M.P.H. WIND U=.313
E-3" CONCRETE, 10" AIR SPACE, METAL LATH AND PLASTER U = .365

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***Graduate Student, University of Pittsburgh.

[1] Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet. A.S.H.V.E. JOURNAL SECTION *(Heating, Piping & Air Conditioning)*, Sept., 1934, p. 399.

Presented at the 41st Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, BUFFALO, N. Y., January, 1935.

The practice usually followed in cooling such offices is to place therein a cabinet containing direct expansion cooling coils, supplied with liquid refrigerant from an electrically driven compressor located in some adjoining space. In this study it was necessary to make an accurate determination of the rate of heat extraction from the room by the cooling system. Since the usual cooling practice does not lend itself to such measurements, the arrangement shown in the diagrammatic sketch, Fig. 1, was used. One commercial cooling cabinet of approximately one ton capacity and one condensing unit of approximately two tons capacity were used in connection with each of the three rooms. The three combinations

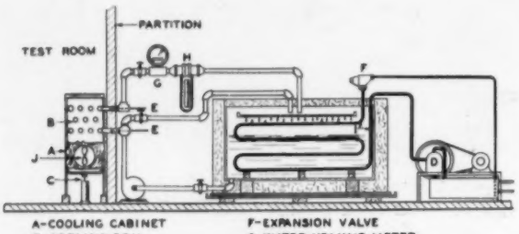

Fig. 1—Diagrammatical drawing of test equipment

were loaned for the purpose by three different manufacturers of such equipment.

Each of the condensing units was connected to a copper, direct expansion cooling coil, partly immersed in water in a tank, 4 ft long by 2 ft high by 2 ft deep. Water was circulated from the bottom of this tank by a small motor-driven centrifugal pump through the cooling coil in the cooling cabinet, then through a calibrated liquid meter of the type used in gasoline stations, and a calibrated orifice flow meter, into a shallow pan with a perforated bottom, from which it returned to the tank by trickling down over the direct expansion cooling coil. Mercury thermometers and compound differential thermocouples, with their sensitive elements located at the middle of the partition, gave the temperature of the water supplied to and returned from the cooling cabinet and the temperature rise. This temperature rise and the quantity of water circulated for any period of time gave an accurate measure of the heat extracted from the room. Both the volume meter and the flow meter were calibrated and frequently checked by weighing the water discharge through them during a given period of time. The volume meter was found accurate to one per cent or better for most rates of water flow. With extremely low rates of water flow it was found to be less accurate, and for such conditions the orifice flow meter gave more satisfactory results. The use of the two instruments served to give a continuous check on each other.

The best cooling results were obtained by keeping the expansion coils in the tank about one-half immersed in water and allowing the water from the perforated pan to trickle down over the upper portion of the tubes as in a Baudelot cooler. The water tank, water circulating pipes, and the exposed expansion valve and refrigerated pipe lines were all lagged with rock-wool insulation so as to conserve cooling. The water supply tank was of sufficient capacity to carry the cooling load for an hour or more at most cooling rates, making it possible to operate the compressor intermittently. With this arrangement, and allowing the formation of ice on parts of the immersed expansion coils, very constant temperature of water supply was maintained, which carried over from closing time of one test to the beginning of the test period the next day.

An observer made the rounds of the three rooms continuously throughout the test period, taking the necessary readings and making the necessary adjustments. The test period began at 7:30 a. m. and closed at 4:00 p. m., Eastern Standard Time (8:30 a. m. to 5:00 p. m., Daylight Saving Time.) Each room was visited at 15 or 20 min intervals. The rate of cooling within the room in order to maintain the desired temperature was manually controlled by the observer on each round by adjusting either the rate of water flow, or to some extent, the rate of air circulation through the cooling cabinet by covering part of the outlet grille with a felt pad. The water supplied to the cabinet usually ranged from 33 to 35 F, while the return water ranged from about 36 to 44 F. The control of the rate of cooling was varied between controlling the rate of water circulation and the rate of air circulation through the cabinet so as to give a water temperature differential of not less than 3 nor more than 10 deg.

The dehumidification rate was determined from the rate of water drip from the cooling cabinet. This naturally did not permit precision measurement, since a large and probably variable quantity of condensed water adhered to the cooling coils and other surfaces of the cabinet before dropping off into the collecting pan, which, having a large water surface area, permitted further variation in rate of water elimination compared to the simultaneous rate of condensation. However, after the system had been in operation for a few hours the rate of water elimination due to dehumidification was markedly constant.

All electrical current used within the room under test passed through a watthour meter which was simultaneously read with the cooling water flow data. The resulting heat supply to the room was calculated and deducted from the rate of heat extracted given by the cold water circulation measurements.

Temperatures within and about the rooms were observed as follows. The representative room temperature or control temperature was observed by a mercury thermometer in the center of the room 30 in. from the floor and shielded from both the cooling cabinet and the window. A wet-bulb thermometer located next to it with a small fan blowing air over both thermometers gave data from which the relative humidity and moisture content of the air in the room could be determined. At the same location in the room plan the temperature of the floor surface, the air temperature 6 in. above the floor, and 6 in. from the ceiling, and the ceiling surface temperature were observed by thermocouples. Wall surface thermocouples likewise gave the temperature of the surface of each of the four room walls at a point aimed to give as nearly as possible a reading representative of that wall. Such points were chosen so as not to be affected by undue radiation or air currents resulting

from sunshine or the operation of the cooling cabinet. Thermocouples or thermometers also gave the temperatures of the air 3 ft outside of the window, the window glass surface, and the temperature of the air in the adjoining rooms and corridor. The constantan leads from all copper-constantan thermocouples in and about each room were carried to a cold junction. Copper leads from the three rooms were carried through a selective switch to a precision potentiometer set-up in a central observation station near room 15.

The wind velocity above the building was indicated by a cup anemometer, located at that point and connected with a counter placed on the 27th floor. The time during which each room was occupied by an observer was noted in order that the heat supplied from this source could be determined.

A mercury thermometer was placed several inches away from the masonry walls in a doorway, 5 ft above the street level, on the Third Avenue or southwestern side of the building, in order to give the air temperature in the street. Thus located, the thermometer had a *view* of the sidewalk, street and masonry buildings across the street, but not of any portion of the sky. Its reading therefore was determined by the street air temperature and radiation from the surfaces in view of it.

In order to determine the moisture content of the outside air, a wet and dry-bulb fan psychrometer was located in a corner room (2701) having a 22 ft exposure with two windows on the northeast, and a 21 ft exposure with one window on the southeast sides of the building. These windows were kept open for some time before, and during the test period. All doors leading to other portions of the building were kept closed, excepting for occasional admittance of the observer. This room was therefore subjected to free exchange of outside air, and although the temperatures indicated by the psychrometer were considerably higher on most occasions than if they had been in space outside the building, the moisture content in grains per pound of dry air calculated from these temperatures should be quite representative of outside air.

In all but a few tests, the value chosen for the control temperature 30 in. above the floor line in the center of the room was that required by THE GUIDE standard based upon outside temperatures. For this purpose the temperature observed outside of the window on the shaded side of the building was used as the outside temperature. An effort was also made to maintain the inside moisture content specified in THE GUIDE standard. While this requirement was usually met, the dry-bulb temperature was considered the more important, and in order to maintain it without undue complication in controlling the dehumidifying load, the indoor relative humidity at times varied, usually resulting in a somewhat lower percentage.

All tests were made with the windows closed, and all but a single test were made with the doors closed, excepting for entrance and exit of the observer. All windows were equipped with inside adjustable Venetian blinds. A single test was made with the door open a part of the time in order to determine the increase in cooling load for this condition. Two tests were made, one at a relatively high temperature and one at a relatively low temperature, with several persons in the room, in order to determine the cooling load resulting from occupancy, and as a practical check on the rate of heat and moisture given off by the human body at various atmospheric conditions as determined by the A.S.H.V.E. Research Laboratory[2]. One test was made with electric lights on near the ceiling in order to determine the degree to which the resulting energy dissipation was reflected in the cooling load.

Test Results

Thirty successful tests were made during the summer on one or more of the three rooms. The calculated results are given in Table 2. The values for total heat extracted were obtained by plotting the instantaneous rates of heat extraction determined from the temperature rise and quantity of water circulated, and integrating this curve for the entire period of the test. The average rate of heat extraction from the rooms was obtained by dividing this total by the number of hours in the test. The total heat extracted for the entire test is compared with the total calculated heat gain for the same period and with the solar radiation gain.

The total calculated heat gain is taken as the sum of the transmission, solar radiation, dehumidification, and infiltration gains, and the calculated heat dissipation by the observer while in the room. The calculated transmission gain is the sum of the calculated hourly heat transfer rates from each of the walls, ceiling and floor, at their prevailing inside surface temperatures and the simultaneously determined air temperature, using a film conductance coefficient of 1.65. For the side walls, an air temperature representing the mean from ceiling to floor was used. For the ceiling and floor, the observed air temperatures 6 in. from these surfaces were used. Also in the transmission gain was included the heat transfer through the window calculated from the hourly observed air temperature 3 ft outside of the window and the window glass temperature.

The solar radiation gain when the Venetian blinds were not lowered, or when the window was bare, was taken as the total solar energy incident upon the glass surface. This was determined by computing the hourly cross-sectional area of radiation normal to the direction of the sun's rays striking the window and multiplying this by the simultaneous solar intensity normal to the direction of radiation observed by the pyrheliometer. The total solar radiation impinging upon the glass at any instant is given by the formula, developed in an earlier Laboratory report[3].

$$E = P\ \left(\sin\ \left[\cos^{-1}\sqrt{\{\cos\alpha\cos(90^\circ-\gamma+\theta)\}^2+\sin^2\alpha}\right]\right)$$
$$(wh - hL\tan(\gamma-\theta) - wL\tan\alpha + L^2\tan(\gamma-\theta)\tan\alpha) \quad [1]$$

where:

E = solar intensity, in Btu per hour, on the window, h feet high, w feet wide, set back a distance L feet from the outside wall surface

P = solar intensity, in Btu per square foot per hour, normal to the direction of radiation at the instant considered.

α = altitude angle of the sun at the instant considered.

γ = angle between north and the direction the window faces.

[2]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p 245.

[3]Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet and J. L. Blackshaw. A.S.H.V.E. JOURNAL SECTION *(Heating, Piping & Air Conditioning)*, Feb., 1934, p. 67.

Table 2—Analysis of Cooling Load

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39
						ROOM 3											ROOM 15											ROOM 18										
						Heat Extracted				Calculated Heat Gain For Entire Test In 100 Btu					Percentages		Heat Extracted				Calculated Heat Gain For Entire Test In 100 Btu					Percentages		Heat Extracted				Calculated Heat Gain For Entire Test In 100 Btu					Percentages	
Test Number	Date	Max. Outside Temp. F	Mean Outside Temp. F	Min. Outside Temp. F	Time Test Started Room 18	100 Btu Entire Test	100 Btu Av. Per Hour	Max. For One Hour: 100 Btu	Max. For One Hour: Beginning Hrs After Start Of Test	Transmission	Solar Radiation	Latent Heat	Infiltration	Observer	Total Calculated / Total Extracted %	Solar Radiation / Total Extracted %	100 Btu Entire Test	100 Btu Av. Per Hour	Max. For One Hour: 100 Btu	Max. For One Hour: Beginning Hrs After Start Of Test	Transmission	Solar Radiation	Latent Heat	Infiltration	Observer	Total Calculated / Total Extracted %	Solar Radiation / Total Extracted %	100 Btu Entire Test	100 Btu Av. Per Hour	Max. For One Hour: 100 Btu	Max. For One Hour: Beginning Hrs After Start Of Test	Transmission	Solar Radiation	Latent Heat	Infiltration	Observer	Total Calculated / Total Extracted %	Solar Radiation / Total Extracted %
1*	July 19	93	78	62	7:30	260	30	37	0	220	23	16	8	2	103	9												370	42	55	0	240	78	30	16	3	99	21
3	26	96	83	70	7:30	886	111	146	2.5	810	37	11	9	3	98	4												638	77	151	1.25	536	129	17	26	4	112	20
4*	27	87	79	71	7:25	552	74	125	0	517	40	31			107	7												623	86	122	4.5	536	89	40			104	13
5	30	88	75	62	7:41	631	76	117	0	564	65	7	6	4	102	10												646	78	95	0	442	157	65	38	5	109	24
6*	31	81	73	65	7:40	437	58	101	2.5	425	9	17	0	3	104	2												509	66	81	1.0	367	120	24	0	3	101	24
7*	Aug. 1	86	72	59	7:21	431	51	78	0	374	34	0	13	4	99	8												335	52	74	0.5	284	40	0	9	2	100	12
8*	2	79	72	64	7:20	133	67	87	0	133	4	0			103	3												132	66	73	0	130	5	0			102	4
9	7	87	73	59	7:38	255	33	66	0	247	27	0			107	11	215	28	65	0	188	41	0			107	19	282	36	66	0	198	157	0			126	56
10*	8	88	78	68	7:25	259	31	36	0	202	21	17	12	3	99	8	318	39	44	0	268	10	24	15	3	101	3	385	44	57	0	248	81	32	18	3	99	21
11	9	90	80	70	7:25	498	60	62	0	439	6	55	5	2	102	1	424	53	63	0	315	52	55	8	3	102	12	566	69	72	7.0	377	148	73	14	2	108	26
12	10	88	81	74	7:30	536	67	93	0	404	30	96	24	2	104	6	413	52	69	0	327	35	56	17	2	106	8	571	69	95	0	397	140	70	20	2	110	25
13*	13	87	76	65	7:53	315	41	63	0	239	6	63	6	1	100	2	337	43	67	0	247	27	54	7	2	100	8	432	56	67	1.0	247	112	61	6	2	99	26
14	14	85	78	70	7:37	358	49	69	0	252	30	77	27	2	108	8	339	44	61	0	270	59	45	11	2	114	17	425	55	77	0	281	182	44	11	2	122	43
15	15	82	74	66	7:46	409	53	90	0	400	6	68			116	1	300	39	59	0	231	76	44			117	25	432	56	89	0	278	172	64			119	40
16*	16	75	70	66	7:41	396	50	74	0	363	3	73	14	2	115	1	329	41	63	0	288	24	29	5	2	106	7	447	56	77	0	332	83	68	8	1	110	19
17*	17	85	75	65	7:53	244	33	75	0	188	10	45	16	2	107	4	230	30	57	.25	169	23	28	8	2	100	10	300	43	54	0	163	78	40	16	1	99	26
18	20	74	66	58	7:35	393	48	84	0	311	46	44			102	12	369	45	70	0	359	13	17			105	4											
20	23	79	70	62	7:34	239	31	48	0	206	5	24			96	2	158	22	34	0	139	15	9			103	9	332	42	64	0	280	106	38			128	32
21	27	78	66	55	7:33	173	23	41	0	132	29	7	8	2	103	17	136	18	26	0	115	33	3	6	2	117	24	324	42	67	0	223	203	1	2	2	133	63
22	28	70	62	53	7:44	142	19	37	4.5	141	12	2			109	8												309	40	52	0	220	146	14			123	47
23	29	67	58	49	7:35																							236	28	41	0	119	231	0			148	98
24*	30	72	59	46	11:47																							265	71	35	2.5	113	171	5			109	65
25	31	75	64	54	7:30																							204	28	35	6.5	81	233	7	2	2	159	114
26*	Sept. 4	77	68	58	7:35	270	35	50	7.5	243	12	23			103	4	306	37	54	0	286	7	27			105	2	445	54	67	0	397	61	40			112	14
27	5	75	66	57	7:45	166	21	39	0	151	12	4			101	7												290	35	45	7.0	142	294	5			152	101
28	6	80	68	56	7:45	81	14	23	2.25	60	28	1			110	35	92	25	23	0	89	23	1			123	25	319	40	49	2.5	179	246	16			138	77
29	7	82	70	59	7:37	118	15	22	0	108	17	1			107	14	117	15	20	1.0	99	20	15			115	17	322	42	49	0	239	174	3			129	54
30*	10	81	70	58	7:45	216	28	45	0	191	9	20			102	4	198	26	40	125	177	17	16			106	9	406	52	63	0	250	144	29			104	35
32	12	83	74	66	8:00												277	36	41	0	256	20	24			109	7	372	48	56	0	243	203	23			126	55
33	13	74	70	66	7:45	631	87	112	5.0	431	4	186			98	1	270	37	57	0	210	28	45			105	10	399	54	76	0	222	216	47			123	55

* Inside Venetian Blind Closed

θ = azimuth angle of sun (angle between sun and the north) at the instant considered.

This paper presented justification for calculating the total heat gain through a window, by solar radiation and transmission, as the sum of the radiation incident upon the glass and the transfer (either positive or negative) from the outside air to the glass. It was again observed in this study that with intense radiation the glass temperature might be higher than both the inside and outside air temperatures, resulting in negative transfer from the outside air to the glass. This is of course accounted for by the absorption of solar radiation by the glass, resulting in a rise in glass temperature and dissipation of the absorbed energy to both the inside and outside air.

Tests made with Venetian blinds are indicated by a footnote in Table 2. For these tests the Venetian blinds were lowered and the slats set at 45 deg with the horizontal, thus excluding all direct radiation into the room through the blind. For these conditions, the sum of heat gain through the window by direct radiation and transmission calculated for a bare window was reduced to 58 per cent in making the heat balance. This is in accordance with earlier Laboratory findings[3] for a closed Venetian blind on the inside of the window.

The dehumidification load was calculated from the rate of dehumidification drippage from the cooling unit. Since considerable lag existed between the instantaneous dehumidification rate and the drippage, the average drippage rate during the later part of the test was used, and this rate was assumed to have been constant throughout the day. The infiltration rate in cubic feet per hour is given by Q in the formula

$$Q = \frac{a\ e}{b - c} \quad \text{.....[2]}$$

where:

a = grains of moisture removed per hour;
b = grains of moisture per pound of dry air outside;
c = grains of moisture per pound of dry air inside;
e = cubic feet per pound of dry air 50 per cent saturated with moisture;

and the sensible heat gain by infiltration in Btu per hour is given by H in the formula

$$H = 0.018\ Q\ (t_o - t) \quad \text{.....[3]}$$

where:

Q = infiltration rate in cubic feet per hour;
t_o = average surrounding outside temperature;
t = room temperature.

This rate was determined for the later part of the test when the dehumidification drippage rate was most constant and assumed to be constant throughout the day.

In this connection it is of interest to note the magnitude of the sensible and latent cooling requirements for the air contained within the rooms for a change in conditions from 88 F and 120 grains of moisture content at the time of the beginning of the test to 77.4 F and 70 grains of moisture content. The heat removal for this change amounted to 368, 204 and 383 Btu of sensible heat, and 1,039, 577 and 1,082 Btu of latent heat, for rooms 3, 15 and 18 respectively.

The heat dissipated to the room by the observer was determined from the total of the various intervals of time when he was in the room and the rate of heat dissipation, based upon earlier Laboratory findings[2]. Theoretically, some of this had already been included in the dehumidification load. However, as shown by the values in Table 2, the total was barely significant and this correction was not made.

The results of a typical test, No. 12, made on August 9, are plotted in Fig. 2. In the upper portion of the chart are plotted the required temperature on the particular day, based upon the outside temperature and THE GUIDE standard; the control temperature of each of the three rooms; the inside surface temperature of one of the walls of room 3; the temperature in the corridor of the building; the weather bureau temperature; and the temperature observed 3 ft outside of the window of each of the three rooms. In the lower three sections of the chart the rate of heat extraction from the rooms, the calculated rate of heat transfer into the rooms through the walls, floor, ceiling and window, the solar radiation gain to the rooms through the windows, and the rate of latent heat extraction are plotted for each of the three rooms. The moisture content of the air in the room, expressed in grains per pound of dry air, and the resulting relative humidity, expressed in per cent, are also plot-

[2]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245.

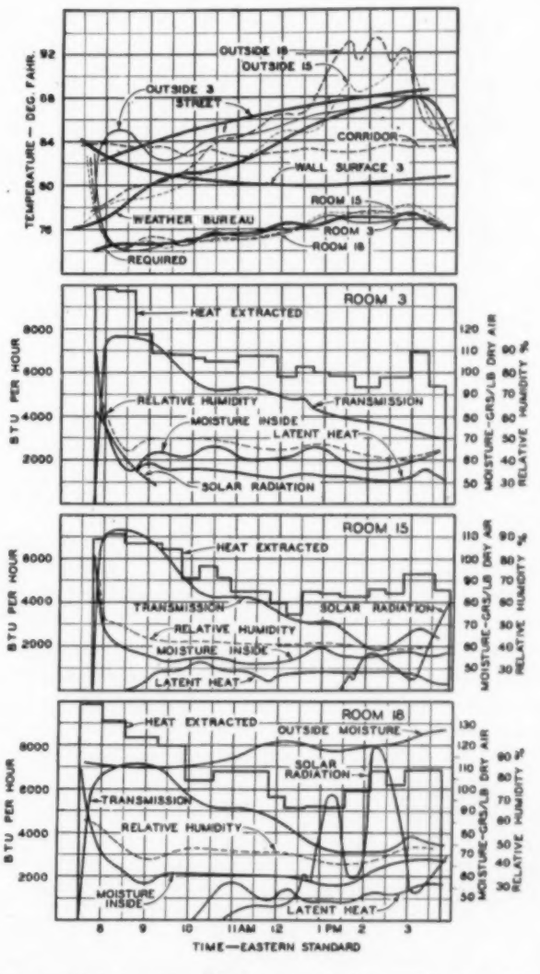

Fig. 2—Relationships during a typical test—Number 12, August 10

[3]Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet and J. L. Blackshaw. A.S.H.V.E. JOURNAL SECTION (*Heating, Piping & Air Conditioning*), Feb., 1934, p. 67.

ted for each room. In the bottom section the moisture content of the outside air is plotted.

The curves representing the rate of heat extraction from each room should equal the sum of the curves representing the transmission through the walls, floor, ceil-

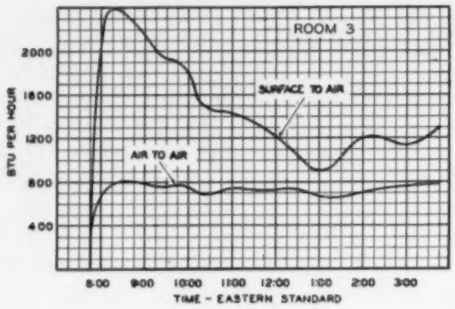

Fig. 3—Calculated heat transfer based on surface to air, and air to air temperatures, for left partition, Room 3, Test 12, showing the effect of heat capacity of the wall

ing and glass, the solar radiation through the window, the latent heat removed by dehumidification, and the sensible heat gain to the rooms by infiltration and by dissipation of heat by the observer. The last two values are small in value and are not plotted. For this test they represented respectively 4.5 and 0.4 per cent of the total heat extracted for the entire test period.

The Effect of Heat Capacity on the Cooling Load

One of the most striking points brought out by the study and demonstrated by the curves in Fig. 2 is the fact that the cooling loads during the early morning were larger than during the later part of the day, resulting from the high rate of heat transfer into the rooms, which reaches a peak soon after the beginning of the test and then gradually decreases as shown by the curves. This is accounted for by the large temperature difference between the inside wall surface and the air in the room during the morning, as shown by the difference between the temperature curves for the three rooms and the single curve, Fig. 2, giving the temperature of one wall surface in room 3. All the inside surface temperature curves plotted show this same characteristic. Due to the small heat capacity of the air within the room and the capacity of the cooling unit, the room air was brought down to the required temperature within 30 min. However, due to the high heat capacity of the walls, the wall surface temperatures did not follow this sudden drop, resulting in a maximum temperature difference between surface and air soon after the beginning of the test: a difference which gradually diminished until the late afternoon.

The calculated rate of heat transfer based upon the surface to air temperature difference for the left wall of room 3 during test 12 is given in Fig. 3. This rate of heat transfer is based upon the entire area of this wall, the difference in temperature between the wall surface and the room air temperature plotted in Fig. 3, and a film conductance coefficient of 1.65. The calculated heat flow for the same wall, based upon the observed air temperatures on the two sides of the partition and a transmittance coefficient, $U = 0.400$, is also plotted for comparison. It will be observed that the surface to air curve gives a much higher rate of heat transfer during the first part of the test and a much greater total heat transfer during the entire test than is indicated by the air to air curve. From the beginning of the test until 10:15 a. m. the surface to air curve gives a total heat transfer of 5,145 Btu, while the air to air curve gives a transfer of 1,828 Btu, or a difference of 3,317 Btu. From the time of the beginning of the test until 2 p. m. the surface to air curve gives a total heat transfer of 8,542, and the air to air curve, a total heat transfer of 3,745, or a difference of 4,797 Btu. The surface temperature curve, Fig. 2, gives a 3.5 deg surface temperature drop for the inside surface of this wall from the beginning of the test till 10:15 a. m., and 4.0 deg till 1 p. m. These temperature drops multiplied by the weight of the wall, 6,855 lb or 36.6 lb per square foot, and an assumed specific heat of 0.23, give heat liberations of 5,518 Btu and 6,307 Btu for the two periods, assuming that the entire temperature drop applied throughout the wall; or 2,759 Btu and 3,153 Btu, assuming only one-half of the temperature drop to apply. The true condition will lie between these two assumptions and will entirely account for the high cooling load during the early part of the day.

Columns 9, 20 and 31 in Table 2 show a maximum rate of heat extraction usually occurring during the first hour of the test, considerably higher than the average rate. An average of all the tests tabulated for each of the rooms gives maximum rates which are 152 per cent, 162 per cent and 130 per cent of the average rate for rooms 3, 15 and 18, respectively.

The high peak in the rate of heat transfer into the room from the surrounding wall surfaces, reached during the morning hours and resulting from the large temperature difference between the required indoor temperature and the wall surface temperature, suggests the possibility of a considerable reduction in the maximum cooling load during these hours by maintaining a somewhat higher indoor cooling temperature in the morning. In order to illustrate this possibility, Test 11 on August 9, plotted in Fig. 4, was made with indoor temperatures about 3 deg higher than the low temperature dictated by the standard requirement. These temperatures are plotted in the upper portion of the chart, together with other pertinent temperatures observed during the test. The effect on the rate of heat transfer from the wall surfaces into the air and the effect on the required rate of heat extraction from the room in order to maintain these higher temperatures are also shown by the curves. The highest portion of the heat transfer curves is diminished, resulting in much flatter curves for the entire day. In fact, this change in procedure resulted in practically constant heat extraction curves for rooms 3 and 18.

Solar Radiation Through Windows

Room 3 received solar radiation in the early morning, which was never great in magnitude and ceased to be effective at times ranging from 10:20 a. m. to 9:37 a. m. during the period July 1 to September 1. Room 18 began to receive solar radiation at times ranging from

10:55 to 9:55 a. m. during the period July 1 to September 1, while room 15 began to receive radiation at times ranging from 1:07 to 1:30 p. m. during the same months. On August 10, the date of Test 12, plotted in Fig. 2, there were intermittent clouds during the afternoon, diminishing considerably the heat gain from solar radiation. However, this gain as plotted for rooms 15 and 18 was sufficient to predict a much greater increase in the cooling load than given by the heat extraction curves. This was one of the surprising facts brought out by the investigation. The study of solar radiation through windows made by the Laboratory a year ago[3] indicated that radiation through a window was immediately available in the air within the room, a conclusion which is not borne out by this study. The reason for the disagreement is apparent in the fact that in the study made a year ago the solar radiation entering the room was allowed to fall on dark gray cheesecloth screens, placed between the window and floor so as to intercept all radiation and also to allow air circulation over the cloth surface. The purpose of the earlier investigation was not so much to determine the speed with which solar energy was absorbed by the air within the room, as it was to determine its rate of entrance through the window. In the present investigation no attempt was made to intercept the solar radiation by any unusual means, and as a result it fell upon the bare floor, which, in the Grant Building, consisted of a 3-in. concrete slab having cemented thereto ¼-in. imitation tile made of a rubber composition or ¼-in. brown linoleum. Since the floor had a very high heat capacity and conductivity, or a high diffusivity, the solar radiation absorbed by it was conducted downward and held in the mass, to be dissipated back to the atmosphere in both the room above and below at a slow rate over a period of 24 hours or longer. The fact that the conductance of the 3-in. concrete slab is 2.42 times greater than the film conductance coefficient from the floor surface to the air insures rapid diffusion of a larger part of the heat throughout the concrete mass. Assuming a concrete density of 150 lb per cubic foot and a specific heat of 0.23, each square foot of the floor would absorb 8½ Btu per square foot per degree temperature rise. Observations made in some of the tests, when solar intensity was high, showed this rise to be as much as 10 deg after three hours of interception of solar radiation. The area of the floor covered by the radiation through the window at any instant varied from about 30 sq ft downwards, but since the area affected did not remain stationary, but moved across the floor with change in direction of the sun, a much larger area of the floor was affected. The variable nature of the factors involved do not permit an accurate determination of the heat thus absorbed for use in making a heat balance, but it is quite probable that absorption by the floor accounted for the fact that the heat extraction curves do not rise in proportion with the solar radiation curves.

An inspection of the percentages, which the calculated heat gain by solar radiation and the entire calculated heat gain are of the total heat extracted for the different tests in Table 2 when the Venetian blinds were not closed, shows satisfactory heat balances for most tests when the solar radiation gain to any room was small, but unsatisfactory heat balances when the solar radiation gain was large. The heat balance was also satisfactory in most tests when the Venetian blinds were closed, providing the heat gain by solar radiation and transfer through the window was assumed to be 58 per cent of the total calculated for the bare window. For other tests, the excess over 100 per cent which the total calculated heat gain was of the total heat extraction is approximately equal to one-half of the per cent which the calculated solar radiation gain was of the total heat extraction. As an example, in test 23, Table 2, room 18, the calculated solar radiation gain given in column 39 was 98 per cent of the total heat extracted, while the total calculated heat gain, column 38, is 148 per cent, instead of 100 per cent of the total extracted. The data indicate that particularly for rooms 15 and 18, receiving considerable solar radiation in the afternoon, only approximately half of the heat entering the unshaded windows by radiation was immediately taken up by the air, the remainder being retained by the floor structure. The tests made with the Venetian blinds closed indicated that for this condition the 58 per cent of the calculated heat gain for a plain window estimated to enter the room did become effective in the cooling load.

It should be observed in this connection that floors of

[3] Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet and J. L. Blackshaw. A.S.H.V.E. Journal Section (*Heating, Piping & Air Conditioning*), Feb., 1934, p. 67.

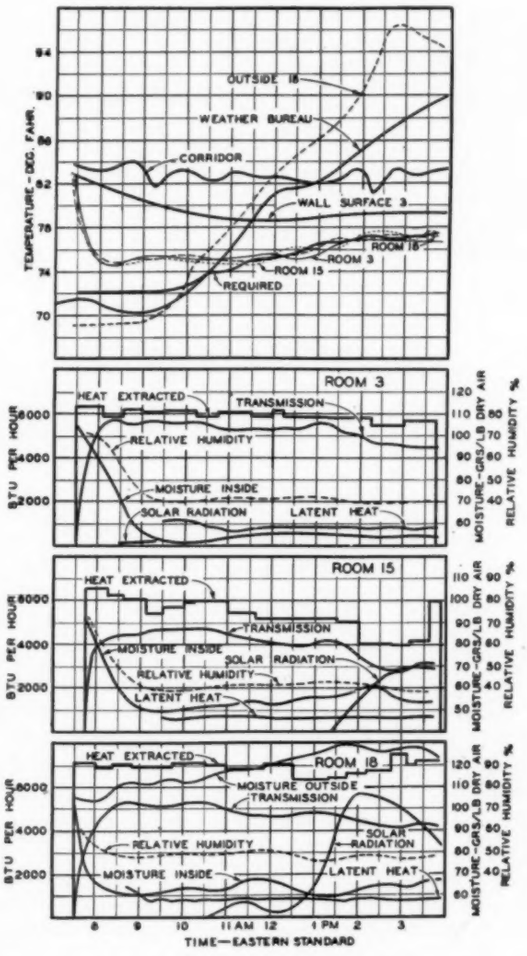

Fig. 4—Relationships during special test, Number 11, August 9, when standard required air temperature for cooling was not maintained

wood construction with lower conductivity and heat capacity, floors with heavy insulating rug coverings, or the presence of furniture so placed as to intercept the radiation, which was lacking in the rooms studied, would minimize the absorption of radiation and therefore the application of the results here reported. Also, that with a high rate of absorption by the floor, the lag between the solar intensity and rate of heat absorption by the air

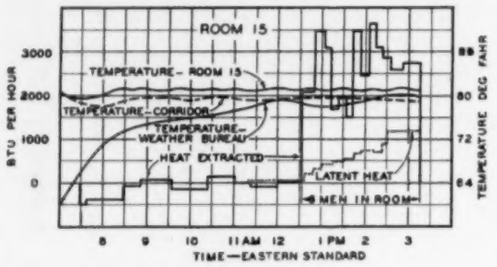

Fig. 5—Effect of occupants on the sensible and latent heat cooling load

becomes a more important factor and, hence, greater solar radiation gain during the earlier part of the day may play a greater part in the maximum and total cooling load. As an example, a south window may give results which could not be predicted with sufficient accuracy from the results of this study, as was assumed in planning the test. This phase of the subject requires further study.

Cooling Load Resulting from Occupants

Two tests were made with a number of occupants in a room in order to determine the resulting increase in the cooling and dehumidifying loads. The results of one of these tests made on September 11 are plotted in Fig. 5. This test was made at a room temperature of 81 F, and in order to minimize the errors resulting from heat transfer into the room it was made on a day when the cooling load from sources other than the occupants was small. The cooling unit was operated at a low capacity throughout the period of the test prior to the time the occupants entered the room in order to keep it in equilibrium with the room conditions. Because the temperature maintained was high compared to the outside conditions, considerable heating had to be supplied before the men entered in order to maintain equilibrium, resulting in negative heat extraction or heating during a part of this test.

Immediately upon the entrance of the 6 men into the room, a great increase of cooling was required in order to maintain the room temperature. This increase averaged 2,618 Btu per hour for the 2¾ hours of occupancy. The dehumidification rate as measured by the collection of condensation increased throughout the greater part of the period of occupancy to a constant rate of 1,150 Btu per hour above the unoccupied rate. The calculated rate of sensible and latent heat dissipation from the 6 men, based upon their weight and height and the earlier findings of the Research Laboratory,[2] was 1,340 and 1,181 Btu per hour, respectively, or a combined rate of 2,521. Considering the crudeness of the test as a calorimeter determination, the results check the laboratory data very satisfactorily.

A similar test was made at an earlier date when the entire cooling load was larger but when a lower temperature of 72 F was maintained in the room. While the results of this test did not check out quite as closely as the one plotted in Fig. 5, it also served to indicate the practical application of the Laboratory's body heat loss data, particularly, that for the lower dry-bulb temperature a larger percentage of the total heat dissipation is sensible.

Effect of Lights on the Cooling Load

In order to demonstrate the extent to which electrical energy used for artificial illumination, with the lights at the ceiling, increased the cooling load for individual office cooling, the test plotted in Fig. 6 was made on a day of moderate outside temperature, with a room temperature of 72 F maintained during the test period. After 4 hours of cooling, when equilibrium had been

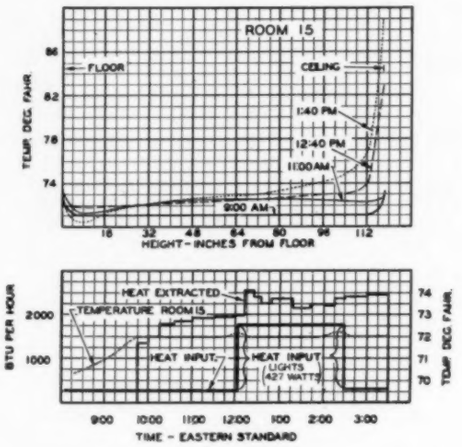

Fig. 6—Effect of artificial illumination on the cooling load

established, two electric lights of 200 w rating each and 427 w combined actual measured electric load were turned on. The combined rate at which electrical energy was supplied to the room for operating the psychrometer fan and the cooling cabinet, before the lights were turned on, and the combined load, after they were turned on, are plotted in Btu. The rate of heat extraction for the entire period of the test is plotted and shows a slowly rising cooling rate required to maintain equilibrium, which reached 1,975 Btu per hour before the lights were turned on, and an average of 2,320 Btu per hour for the period during which the lights were on, or an increase of 345 Btu per hour. It is probable that some of this increased cooling load resulted from increased heat gain to the room, as shown by the upward slope of the heat extracted curve before the lights were turned on. However, even without this consideration, the test demonstrates that only a small part of the

[2]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245.

energy liberated by lights located at the ceiling affects the cooling load for an individual room cooled over a relatively short day. It should be pointed out, however, that this will not necessarily apply in an office building entirely cooled, particularly if the cooling is applied over a larger part of the 24 hours, since the heat stored temporarily in the ceiling and upper floor structure must eventually be accounted for. Temperature gradient curves from floor to ceiling during the period of this test, as plotted in the upper part of the chart, show clearly why the energy dissipated from the lights does not appear in the cooling load. The air heated by the lights stratifies very closely to the ceiling, raising the temperature of the structure, and only serves to raise the temperature of the lower layers of the air a small amount after a considerable lapse of time.

It should be observed that the 427 w for the small room, or 3.9 w per square foot of floor area used, represents a great excess over normal illumination. It was used in this case in order to magnify its measurement. Each of the two light bulbs of the Mazda type were placed in a separate fixture as shown in Fig. 7. The two fixtures were about 3 ft apart, and their closeness to the ceiling accounts for the low rate of heat absorption by the air. A low hanging fixture would no doubt give a much higher rate of heat gain to the air and thus increase the cooling load. Also a cooling unit, giving an air delivery resulting in a positive air current over the bowl and between it and the ceiling, would have the same effect.

Effect of an Open Door on the Cooling Load

Much speculation is met with in estimating cooling requirements of offices and small stores concerning the increase in cooling load resulting from frequent opening of doors. In order to indicate the magnitude of this factor, the test plotted in Fig. 8 was made on September

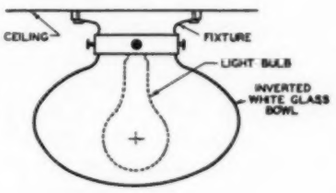

Fig. 7—Electric fixture with inverted white glass bowl

12. In this test the door was kept closed until 1 p. m., when it was opened half way, or to an angle of 45 deg. The rate of heat extraction immediately increased from 5,550 Btu to 10,850 Btu, or by 5,300 Btu per hour. The latent heat load increased from 500 Btu per hour to 3,300, and the infiltration load, calculated from the inside and outside moisture content and the dehumidification drippage, increased from 293 Btu per hour to 2,140 Btu per hour. This represents an infiltration rate of 0.86 air changes per hour for the room before the door was open, and 6.2 air changes with the door open. For this test a definite temperature difference of 10 deg between the corridor into which the door was open and the room tested was maintained, this being assumed to be a representative cooling difference. The dimensions of the door were 83½ in. by 37 in. In applying the results of this test to the estimation of the effect of an open door on the cooling load of a given space, the factors which must be estimated are the difference in temperature and moisture content of the air between the cooled space and the space into which the door is opened, and the length of time per hour during which the door is assumed to be open. The resulting increase in cooling load should be independent of the wall transmission and

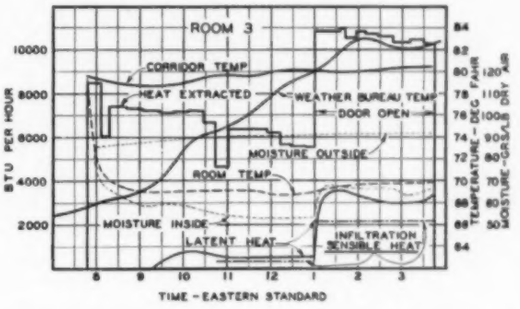

Fig. 8—Effect of opening a door on the cooling load

solar radiation loads. It should be pointed out that the cooling load will not be directly proportional to the air temperature differences on the two sides of the door, but more nearly proportional to the square of this difference, since it will be directly proportional to not only the temperature difference but also to the volume of air change which in itself should increase in almost direct proportion to the temperature difference.

Temperatures Affecting Cooling Requirements

The inter-relationships of the temperatures plotted in Fig. 9 for the period July 15 to August 31 are of interest. The small space allowed for each day is made to represent the period from 6 a. m. to 4 p. m., Eastern Standard Time. The weather bureau temperature curves show the entire 24-hour range reported. The time of the maximum and minimum are located approximately by the upper and lower terminals of these curves. Usually, the minimum reported was around 6 a. m.; if it was earlier, the lowest end of the curve was started slightly to the left of the 6 a. m. ordinate or in the space for the previous day, and if considerably after 6 a. m. it was located slightly to the right. Likewise, the time of the maximum was indicated as coming either before or after 4 p. m. Eastern Standard Time. The shape of the curve shows approximately the rate of temperature variation during the day. The temperature of the air outside of the window of room 18 was always observed in the shade at the beginning of the period, and in the sun later, if it was a clear day. The corridor air temperature was observed on a mercury thermometer located in the corridor near room 18. The circular points give the temperature observed for the left partition of room 3 just before cooling was turned on in the morning.

The relation of the corridor temperature and its daily range and the partition surface temperature in room 3 to the weather bureau mean temperature and temperature range is of particular interest. Over a several day period of rising temperature the corridor temperature

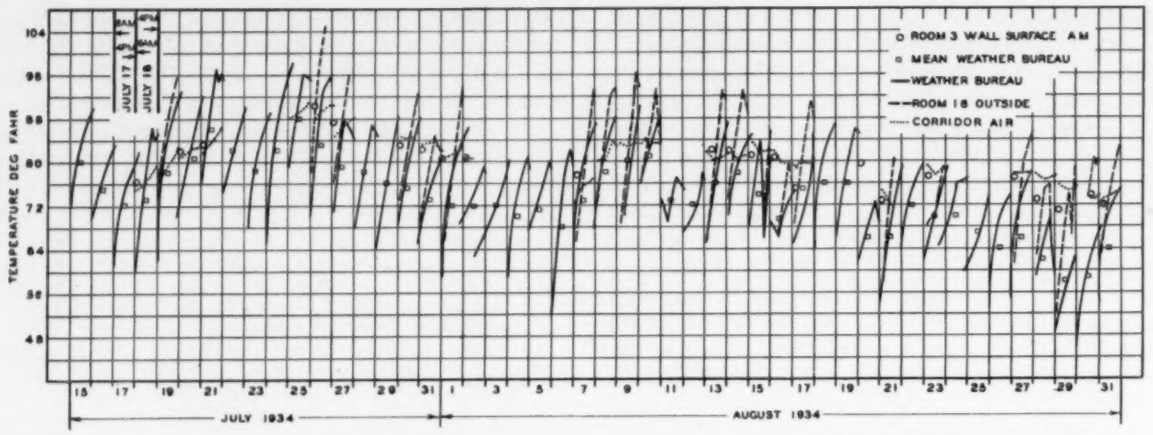

Fig. 9—Variation in temperatures reported by weather bureau, and observed in corridor of building, 3 ft outside of window 18, and for the surface of left partition in Room 3

and the partition temperature lag behind the weather bureau mean temperature, while over a period of declining weather bureau temperatures the reverse is true. For the entire period, July 15 to August 31, the corridor mean temperature and the partition temperature in the early morning average 7.4 and 6.8 deg higher respectively, than the weather bureau mean temperature. This difference must be accounted for by the dissipation of heat within the building from machinery, electric lighting, occupants, and other sources. The small daily temperature range of the air in the corridor, rarely increasing by more than 4 deg during the day demonstrates the great heat capacity of the building structure and the marked tendency of the air within the building to follow the temperature of the structure. This is further demonstrated by the close agreement among the room temperatures, the corridor air temperature and the partition surface temperature in the morning, showing that the heat extraction from a single room during a day was returned during the following 16 hours so as to raise its temperature to practically the same level as observed in the uncooled rooms. In a few instances the windows of the test rooms were left open during the night after a cooling test. This had the maximum effect of reducing the air temperature difference between the outside and the test room, as observed before test in the morning, by one quarter of the difference observed for unconditioned rooms with their windows closed.

The temperature indicated by the thermometer near the street level outside the building bore an interesting relation to the temperatures plotted in Fig. 9. Its temperature range was always considerably less than the temperature outside of the window in room 18 and that reported by the weather bureau. The low temperature in the morning was usually a little above the weather bureau mean, while the high recorded in the late afternoon was usually between the high recorded for the weather bureau and that observed outside the window in 18. The location of this thermometer as described undoubtedly had much to do with the reduced range as well as the magnitude of its readings. While it does not represent general sidewalk conditions on a hot day, it probably does so as well as any other location which might be chosen. More open spaces in the street subjected to greater air circulation and view of the sky would probably give a more extended range.

Estimating Cooling Requirements

The required indoor cooling temperature, dictated by the outside temperatures, used in most of the tests are shown by the study to result in a maximum cooling load during the first hour in the morning approximately equal to the sum of the calculated heat gain from all the inside wall, ceiling, floor and glass surfaces, for the temperature difference prevailing and the latent and sensible heat load based upon the cooling of one room volume of air. For the conditions under which the study was made, this maximum was unaffected by the higher outside temperatures and the intense solar radiation during the later part of the day, and for these conditions it is apparent that any assumption which raises the required low temperature standard during the early morning will reduce the maximum cooling load approximately in proportion to the reduced number of degrees cooling. The present standard of indoor cooling temperature based upon outside temperature is somewhat ambiguous in that the location of the outside temperature is not more closely specified. Four rather different cooling temperature standards might be chosen in the early morning, based upon either the temperature recorded by the weather bureau, the temperature at the street level, the temperature outside the window of the office cooled, or the corridor temperature of the building. The reason for the temperature standard for indoor cooling is the feeling of contrast or shock experienced by an occupant upon entering or leaving the cooled room. Obviously, a strict consideration of this point of view would eliminate the weather bureau temperature and the temperature outside of the window of the room, and limit it to a consideration of either the street temperature or the building corridor temperature. Probably, use of the corridor temperature or the mean temperature of the building would be by far the better selection, since it determines the immediate contrast experienced by the occupant. This would have the operating advantage of a much smaller temperature range throughout the day, in fact, a range so small that the

mean corridor temperature would serve satisfactorily in fixing the cooling temperature for the entire day. By requiring a somewhat higher temperature during the morning hours it would have the distinct advantage of decreasing the peak load during this time. The effect this would have on a required indoor temperature may be illustrated with reference to tests 12 and 11, plotted in Figs. 2 and 4. Basing the required temperature upon the corridor temperature of the building for these days would give an indoor temperature range of 75.7 F to 76.1 F for test 12, and 75.3 F to 76.2 for test 11, and an estimated reduction of 25 per cent in the maximum rate of heat gain by transmission during the first hour. For test 11, this consideration for outside temperature would give approximately the same conditions under which this special test was made.

In applying the accepted standards for indoor summer cooling, the need for further clarification as regards the recognition of the outside effective temperature becomes apparent. Present practice is based upon the outside dry-bulb temperature only. This is probably because the standards were built up through practical applications where the effect of moisture content of the outside atmosphere could not be easily recognized. These standards are also difficult of application, since they allow only one moisture content, fixed by a constant dew point of 57 F, for any indoor temperature. Further study will undoubtedly demonstrate the practicability of allowing some range or latitude in accepted moisture content, provided the proper effective temperature determined by the outside effective temperature is maintained, thus removing an apparently unnecessary restriction in the application of summer cooling. Great difficulty was experienced in operating the cooling units so as to maintain both the required dry and wet-bulb temperatures. When these requirements could not be met, the required dry-bulb temperature was maintained, and the wet-bulb temperature was allowed to vary slightly, but in no case so as to give more than one degree variation in the specified effective temperature.

It is of interest to apply the data to the estimation of the maximum hourly cooling load required for the three rooms for design purposes. Since the corridor temperature of a proposed building in a given city cannot be predetermined, it is desirable to base the estimated heating requirement upon some function of the weather bureau data. Inspection of temperature ranges given in Fig. 9 shows that at the peak of a high temperature wave, the mean weather bureau temperature tends to reach or slightly surpass the building corridor temperature. As an example, on July 21, the weather bureau mean temperature reached a higher value than the corridor temperature, and again on July 25 the weather bureau mean temperature rose to equal the lower limit of the building corridor range.

The four hot days of the entire season were: July 25, with a minimum temperature of 79 F, a maximum temperature of 96 F, and a mean temperature of 88 F; July 21, with a minimum temperature of 76 F, a maximum temperature of 97 F, and a mean temperature of 86 F; June 28, with a minimum temperature of 72 F, a maximum temperature of 95 F, and a mean temperature of 84 F; and July 26, with a minimum temperature of 70 F, a maximum temperature of 96 F, and a mean temperature of 83 F. It is of interest to note that during the past summer season in Pittsburgh there occurred 9 days in June, 9 days in July and 2 days in August, with weather bureau mean temperatures of 80 F or higher.

Following the practice of heating, it would be logical to base cooling design on the average of several maximum mean temperatures occurring over the past several years. Since such records are not easily accessible, it may be logical to base our reasoning on the highest mean temperature occurring during the past summer season, or 88 F, which has been shown to agree closely with the maximum corridor temperature for the summer. This high corridor temperature also occurred on July 25 and ranged from 88 to 91 F. Accepting 88 F as a standard on which to base summer cooling in Pittsburgh, and THE GUIDE relationship between indoor and outdoor temperatures, an indoor temperature of 77.6 F would be required for such a day, giving a 10.4 deg differential between the assumed building temperature and the air temperature maintained indoors. Multiplying the sum of the surface to air transfer coefficients for all surfaces in room 3, or 1,585 Btu per hour per degree temperature difference for the entire surface, times 10.4, or the assumed temperature difference, gives 16,484 Btu heat extraction from the wall surfaces per hour. A like consideration for rooms 15 and 18 gives 10,826 and 16,796 Btu per hour, respectively.

These values are obviously high, since they are based on the assumption that the maximum temperature difference would pertain throughout the first hour, which is not the case. During the first quarter, to one-half hour, the air temperature within the room fell rapidly to the required standard during which time the sensible and latent heat content of the air in the room above the standard condition was being extracted. At the end of this period the wall surface temperature had already fallen somewhat, and continued to fall during the following hour, so that for no hour was the average temperature difference between the wall surface and the air more than ¾ of the maximum assumed. Hence, a more reasonable figure for the maximum requirement would be ¾ of the maximum values calculated above, or 12,363, 8,119 and 12,597 for rooms 3, 15 and 18, respectively. These values may therefore be considered as the maximum cooling requirement for the three office rooms studied for the given type of construction and exposures, and for a building in the Pittsburgh district. Again inspecting Fig. 9 with respect to the maximum corridor temperatures, it will be observed that July 26 had about the same maximum weather bureau temperature and the same corridor temperature. However, because the minimum for that day was considerably lower, the mean was also lower than for July 25. Since the corridor temperatures on the 26th and 25th were about the same, the maximum cooling loads should also be the same. Data for the 25th are not available, but they are available in Table 2 for the 26th. On this day the maximum cooling load for any hour for room 3 was 14,600 Btu per hour, and for room 18, 15,100 Btu per hour. It will be noted in this connection that the maximum rate did not occur during the first hour in this test, for the reason that trouble was experienced in obtaining cold enough water. As a result, the required temperature was approached gradually over the first few hours of this test.

The maximum cooling loads for the 26th as given in Table 2 are higher than our present assumed requirements for the reason that lower room temperatures based upon the outside temperature were maintained during the morning. The application of the smaller temperature range here considered would have brought the values of 14,600 for room 3 and 15,100 for room 18 down at least 20 per cent or less than the estimated values of 12,363 and 12,597, respectively. In other words, these cooling rates would have quite satisfactorily cooled the two rooms to 77.6 F on July 26. Hence, it may be proposed as a result of this study that the maximum cooling requirements for a single office entirely surrounded by masonry walls, floor and ceiling may be given by the formula:

$$H = A f_i (t_s - t_a) \qquad \text{.....[4]}$$

Where:

H = the maximum cooling requirement for any hour.

A = total area of wall, floor and ceiling surfaces of the room in square feet.

f_i = the inside film conductance coefficient, or 1.65 Btu per hour per square foot per degree temperature difference between the wall surface and the air.

t_a = the required indoor cooling temperature for t_s, the highest mean weather bureau temperature assumed for the locality, and also assumed to be equal to the highest wall surface temperatures in the morning before cooling is applied.

In presenting this suggested method of estimating the maximum cooling load it should be strongly emphasized that it is only to be applied to: (1) single offices located in large office buildings and cooled for approximately 8 hours per day; (2) offices entirely surrounded by masonry walls at least as heavy as plastered hollow tile; (3) to offices with windows equipped with inside Venetian blinds or other provisions which will reduce the absorption by the air of the solar radiation through the window to approximately one-half that entering a bare window; and (4) to rooms with other than southern and southeastern exposures. As pointed out, it should be further emphasized that because of the unforseen results of the study and the resulting choice of rooms not including southern and southeastern exposures, the effect of these variables should receive further study.

Summary and Conclusions

1. The study made in the Grant Building during the past cooling season clearly indicates that with the acceptance of present standards for indoor cooling temperatures the main factor in the maximum cooling load is the high rate of heat transfer from the surface of walls, ceiling and floor to the cooled air in the room, and that the peak requirement takes place soon after the beginning of the cooling period.

2. The outside temperature basis for the present cooling temperature standards is shown to have a large bearing on the maximum cooling rate and a lesser effect on the total cooling load for the day. The acceptance of the building corridor temperature instead of the temperature outside of the building as a basis for determining the required cooling temperature for single offices is shown to have some inherent advantage in giving a more desirable indoor temperature, besides the very material advantage of reducing the peak cooling load.

3. For the conditions of construction and exposure in the Grant Building, the solar radiation load had little effect on the peak cooling load for the day. Approximately the same effect on the total cooling load was had with or without the use of inside Venetian blinds. This was the case in the Grant Building, particularly because the concrete floors absorbed and retained practically the same percentage of the solar radiation entering the window as the closed Venetian blinds would have retarded. In this connection it should be pointed out that other variables including a south or southeastern window, and floors with less heat capacity and higher conductivity, may alter these findings in other buildings. These variables should be studied further.

4. The Research Laboratory's earlier findings on sensible and latent heat dissipation from persons were shown to apply to the cooling of a room.

5. The electrical current used for artificial illumination with lights placed near the ceiling was shown to have little effect on the cooling load in an office cooled during the usual short period.

6. The increase in the cooling load resulting from an open door for a single air condition representing average practice is given.

7. A suggested method for estimating the maximum cooling load for an office, based upon the highest mean weather bureau temperature is presented for limited application to single rooms in large office buildings with masonry walls, and for rooms not having a southeastern or southern exposure.

Acknowledgment

Acknowledgment is due Prof. John A. Dent of the Mechanical Engineering Department of the University of Pittsburgh, for student assistance in analyzing the data. In this connection, theses based on the results of the study were presented to the University as follows: Summer Cooling in a Modern Office Building, by J. E. Helquist; and Some Factors in the Cooling of Rooms in a Modern Office Building, by A. J. Wahl. Acknowledgment is also due the Management of the Grant Building for making the rooms available for the study and for other valuable assistance in making the study a success.

Importance of Climate

Climate affects man in many ways—his housing, clothing, food, occupation, migrations, form of government, and manner of living are all more or less influenced by the climate in which he dwells, says *Tycos.* Much the greater part of man's food is derived either directly or indirectly from the products of the soil, and these are likewise markedly affected by climatic conditions. The principal facts that influence the segregation of crops into restricted areas or zones in different parts of the world are topography, character of soil, climate, and distance from market. Among these, climate is the most fundamental, unalterable, and important, not only in influencing the geographic distribution of the crops that are grown, but also in determining the suitability of the land for agricultural purposes in general. Soil fertility and suitability for cultivation are important factors, but they would avail little were the climate unfavorable. Moreover, the fertility of the soil is largely dependent on climate, operating indirectly through vegetative growth and otherwise.

Insulation and Ventilation of Hospital Operating Rooms to Reduce Summer Temperature

By **R. B. Saxon*** ***(NON-MEMBER)***
Omaha, Nebr.

ALL of the colleges of the University of Nebraska are located at Lincoln except the College of Medicine, which is at Omaha. The College of Medicine operates a general hospital of 250 bed capacity to provide clinical material for the medical and nursing students and to assist in the care of the indigent sick of the state.

Fig. 1 is an air view of the Omaha plant showing the laboratory buildings in front and the hospital building near the center. The Nurses' Home is located at the left of the hospital with the heating plant at the extreme left.

Fig. 1—Air view of the College of Medicine, University of Nebraska. (Arrow indicates location of operating rooms)

Excessive temperatures in the operating rooms in the summer months have always been a problem. It was conceived that, if air from the outside could be brought in at night in sufficient quantities and the rooms protected to some extent by insulating the skylight and ceiling, lower day temperatures might be obtained. A pressure type of fan driven by a ½ hp motor with a capacity of 1500 cfm of free air was used to bring the air in from the outside. Recording thermometers were used to measure the temperatures of the operating rooms; ordinary glass stemmed thermometers were used to measure the wall and ceiling temperatures. A differential thermostat with a mercoid switch controlled the fan to operate only when the outside temperature was lower than the inside temperature. Dry filters were installed in the fan house to clean the air brought in from the outside. A recording voltmeter was used to indicate the time the fan was in operation. Outside temperatures were taken from the U. S. Weather Bureau records. Because of lack of facilities, no attempt was made to determine or control the relative humidity.

The operating rooms are located on the top floor as indicated by the arrow, Fig. 1. Fig. 2 is a plan of the operating floor, and Fig. 3 is a section of the rooms studied, showing the fan house, air ducts and insulation applied. The changes were made and readings taken between June 9 and August 31, 1933. It was decided to ventilate and insulate the east room, leaving the west room unchanged to serve as a control. However, a month later, upon the insistence of the surgical staff, it was necessary to insulate and ventilate the west room in a manner similar to that which had been done in the east room.

Changes in the East Operating Room

The first step was the shading of the skylight (skylight 11x14 ft) with canvas. This seemed to improve conditions somewhat. On June 12, five days later, the skylight was permanently cut off by applying two half-inch layers of insulating wall board on the inside (the conductance, *C*, of the wall board used was 0.34). This almost at once lowered the temperature as indicated in Table 1. On June 22 the blower was installed and started by the night watchman at 3:00 a. m. and stopped at 7:00 a. m. On the same date 4 in. of insulating material (conductance, *C*, 0.28) was applied on the ceiling in the attic. At this time a check of both east and west rooms was made for dust and bacteria. This was

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done under the supervision of the Department of Bacteriology. Glass slides, 3 in. x 1 in., coated with vaseline, were used for dust determinations and agar plates were used for bacteria determinations. The plates were placed on the operating table near the center of the room at 5:00 p. m. and removed the next morning at 7:00 a. m. These tests showed practically no difference in either dust or bacteria between the two rooms, the east room being operated with the fan (filtered air being taken in from the outside) and the west room without a fan (still air); the windows were closed at all times.

sate for the increased volume to be ventilated. On August 1 the differential thermostat to operate the fan was installed, Fig. 4. This device was arranged with a bimetallic element in the room and a similar one outside. The elements were so connected that the tipping of a mercoid switch started the fan when the outside temperature reached a point as low as the inside and stopped the fan when the outside temperature exceeded that of the room. Continuous temperature curves were plotted from June 9 to August 31 showing maximum and minimum temperatures reached during the 24-hour

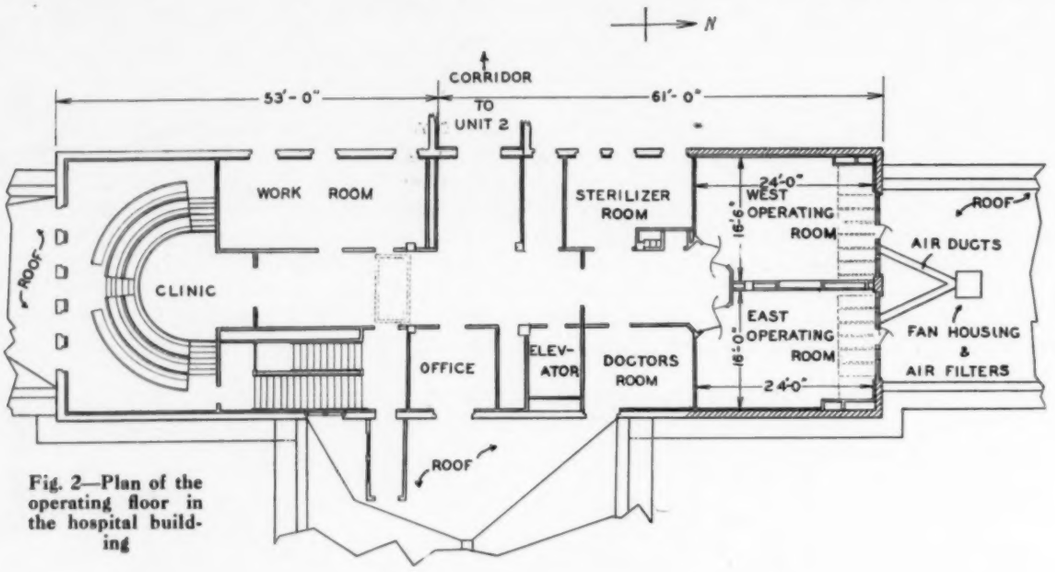


Fig. 2—Plan of the operating floor in the hospital building

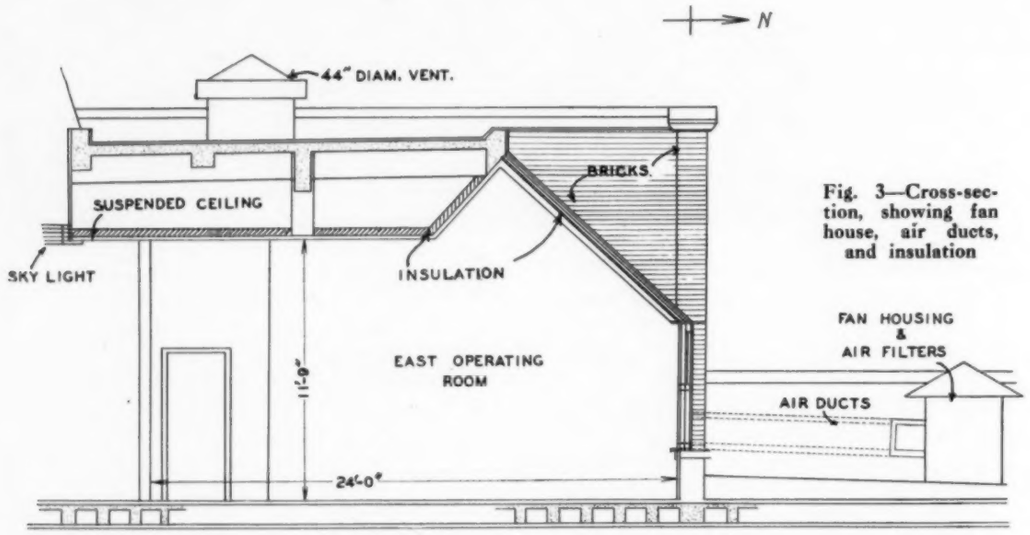


Fig. 3—Cross-section, showing fan house, air ducts, and insulation

Changes in the West Operating Room

On June 30 the skylight was painted on the exterior with two coats of white lead and oil. This reflected some of the heat from the sun and resulted in some improvement of the inside temperature. On July 10 the ceiling of the west operating room was insulated and on July 13 the skylight was closed in a way similar to that explained previously in connection with the east operating room. On July 14 the fan house was rearranged and air ducts were provided to supply air to both rooms. The operating period of the fan was lengthened to compen-

period in each of the east and west rooms and also outside.

To summarize the actual gain in reduced temperatures, Tables 1, 2, 3, 4 and 5 were prepared, selecting days in which the outside maximum temperatures did not vary more than 5 deg. These tables were made at different stages of insulating and ventilating during the summer. In Table 4, the west room does not show as much lowering of temperature as the east room although the changes made were approximately the same. This may be explained in part by the location of

Table 1—Operating Room Temperature Records

(Conditions: W. O. R. unchanged. E. O. R. skylight insulated with two ½ in. layers of wall board.)

	Maximum Outside Temperature	Maximum W. O. R. Temperature	Maximum E. O. R. Temperature	Temperature of W. O. R. Under Outside	Temperature of E. O. R. Under Outside
June 15.....	92	90	85	2	7
June 16.....	94	93	90	1	4
June 17.....	93	95	89	–2	4
June 18.....	95	97	90	–2	5
June 19.....	97	97	93	0	4
June 20.....	93	95	91	–2	2
June 21.....	91	94	88	–3	3
Average	93½	94½	89½	–1	4

Table 2—Operating Room Temperature Records

(Conditions: W. O. R. skylight painted white exterior; ceiling insulated. E. O. R. skylight and ceiling insulated and ventilated with fan manually controlled 4 hours, 3:00 a. m. to 7:00 a. m.)

	Maximum Outside Temperature	Maximum W. O. R. Temperature	Maximum E. O. R. Temperature	Temperature of W. O. R. Under Outside	Temperature of E. O. R. Under Outside
July 5.....	91	90	87	1	4
July 6.....	92	91	87	1	5
July 9.....	90	89	85	1	5
July 10.....	93	91	88	2	5
July 11.....	95	94	91	1	4
July 12.....	93	90	88	3	5
July 13.....	93	93	90	0	3
Average ...	93	89	85	1½	4½

Table 3—Operating Room Temperature Records

(Conditions: Both rooms insulated and ventilated with fan, manually controlled for 6 hours, 1:00 a. m. to 7:00 a. m.)

	Maximum Outside Temperature	Maximum W. O. R. Temperature	Maximum E. O. R. Temperature	Temperature of W. O. R. Under Outside	Temperature of E. O. R. Under Outside
July 20.....	93	89	85	4	8
July 21.....	93	91	88	2	5
July 22.....	95	93	90	2	5
July 27.....	91	89	87	2	4
July 28.....	96	91	89	5	7
July 29.....	98	93	91	5	7
July 30.....	94	93	91	1	3
Average ...	94	91	88½	3	5½

Table 4—Operating Room Temperature Records

(Conditions: Both rooms insulated and ventilated with fan automatically controlled; operating time varying from 6 to 14 hours.)

	Maximum Outside Temperature	Maximum W. O. R. Temperature	Maximum E. O. R. Temperature	Temperature of W. O. R. Under Outside	Temperature of E. O. R. Under Outside
August 7..	94	89	87	5	7
August 10..	92	85	80	7	12
August 11..	93	86	85	7	8
August 15..	94	86	84	8	10
August 16..	92	88	86	4	6
Average ...	93	87	84½	6	8½

Table 5—Operating Room Temperature Records

(Summary of Tables 1, 2, 3 and 4.)

	Average Maximum Outside Temperature	Average Maximum W. O. R. Temperature	Average Maximum E. O. R. Temperature	Average Temperature of W. O. R. Under Outside	Average Temperature of E. O. R. Under Outside
Table 1.....	93½	94½	89½	–1	4
Table 2.....	92½	91	88	1½	4½
Table 3.....	94	91	88½	3	5½
Table 4.....	93	87	84½	6	8½

the sterilizing room adjoining and also because of the west exposure of this room. Thermometers, with the bulb in contact with the wall and insulated from the room, indicated a difference of from 2 to 4 deg between the outer walls of the east and west rooms. Table 5, a summary of the periods represented by Tables 1, 2, 3 and 4, shows a lowering of temperature of 9½ deg. All temperatures given in the tables are maxima reached during the 24-hour period. The minimum temperature reached was 4 to 8 deg below the maximum.

The tables show quite clearly that a considerable lowering of temperature was obtained by the insulating and ventilating. One of the valuable improvements resulting from the closing of the skylights was the change in illumination. Previously the glare of the sun at midday was very uncomfortable on the eyes. Modern surgical lighting units had already replaced the daylight and were used at all times. The contrast of the illumination from a modern operating lamp was much more marked when the direct sunlight was shielded; the effect being more like a spotlight in a theater darkened or with subdued illumination.

The following winter the temperature was maintained between 75 and 78 deg, whereas formerly the range had been from 80 to 85 deg. The surgical and nursing staff all reported this lower temperature to be very comfortable, nor did this show any ill effects upon the patients. Probably the explanation for this is the reduction in radiant heat loss due to the relatively higher temperature of the skylight area and room ceiling. On a theoretical calculation, taking account of the reduction of heat loss due to the insulation and the lowering of room temperature, with a steam cost of forty cents per thousand pounds, a saving of fifteen to twenty dollars per room per heating season was made. This saving in two or three years would pay for the cost of the insulation. No doubt, in many cases, a relatively small investment can be made in insulating materials which will materially aid in summer comfort and be repaid by the fuel savings in winter.

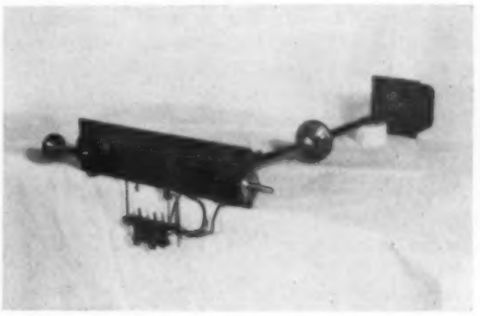


Fig. 4—Differential thermostat with mercoid switch for control of fan

Electric Hot-Water Supply

Heating and electrical engineers will be interested in Hot-Water and Steam Supply by Electricity, by Frank C. Smith, *A.M.I.H.V.E., A.I.E.E.,* recently published by E. & F. N. Spon, Ltd., London.

In its eight chapters, well illustrated, the author has provided information on electric water heaters, covering their installation in large and small buildings, as well as their use in connection with swimming pools. The subjects discussed include temperature measurement, electrical fundamentals, resistance, heating elements and thermostats, heating water by electricity, electric hot-water systems, conversion systems, use of electric energy for swimming baths, and electric steam generation, boilers, immersion heaters, tubular and electrode.

At the Old Mill, Toronto

Toronto Prepares

Members of Ontario Chapter in Toronto are preparing for the Semi-Annual Meeting of 1935, when their fellow-members of the A.S.H.V.E. will assemble for three days of meetings and fun, June 17, 18 and 19th.

The technical program provides for morning sessions from 10:00 to 12:30 and after that the Entertainment Committee is planning for the afternoon and evening hours of members and ladies. For those who golf, there will be golf tournaments and matches; for those who enjoy sight-seeing, auto trips to places of interest have been planned.

A get-together luncheon and an informal supper and dance for members and ladies is scheduled as the concluding feature of the first day's visit. This affair will be held at the Old Mill, a famous and charming spot in the heart of Toronto. The inn and the gardens are delightful and will be fascinating to everyone, as they contain a marvelous collection of things associated with the early history of Canada.

The special program planned for the ladies provides for a boat ride, a visit to the yacht club, and a bridge party, and on Tuesday evening everyone will assemble at the Royal York for a formal dinner dance.

Application has been made for reduced fares from all points in the United States and Canada, and members will receive certificates entitling them to make the round-trip for fare and one-third. Diverse routings are possible and those desiring to make the trip to Toronto a part of their vacation may wish to return via the Thousand Islands, or through Montreal, or by boat on the Great Lakes.

The Committee on Arrangements has held two meetings and, as plans are completed, detailed announcement will come to Society members.

The personnel of the Committee appointed by the Board of Governors of Ontario Chapter and announced by Pres. W. R. Blackhall is: *Chairman,* E. H. Gurney; *Vice-Chairman,* L. L. Anthes; *Finance,* M. F. Thomas; *Publicity,* E. R. Gauley; *Ladies,* H. B. Jenney and Mrs. E. H. Gurney; *Banquet,* Thomas McDonald; *Entertainment,* W. P. Boddington; *Transportation,* M. W. Shears; *Reception,* H. H. Angus; *Registration,* J. S. Paterson; and *Secretary,* H. R. Roth. They want all Society members to come to Toronto for the Semi-Annual Meeting 1935.

Don't forget the dates, June 17, 18 and 19th.

Fourth Heating and Ventilating Exposition Announced

The Fourth International Heating and Ventilating Exposition will be held at Chicago, Ill., in the new International Amphitheatre, January 27-31, 1936. Since its inception in 1930 the Heating and Ventilating Exposition has been held biennially, successively in Philadelphia, Cleveland, and New York.

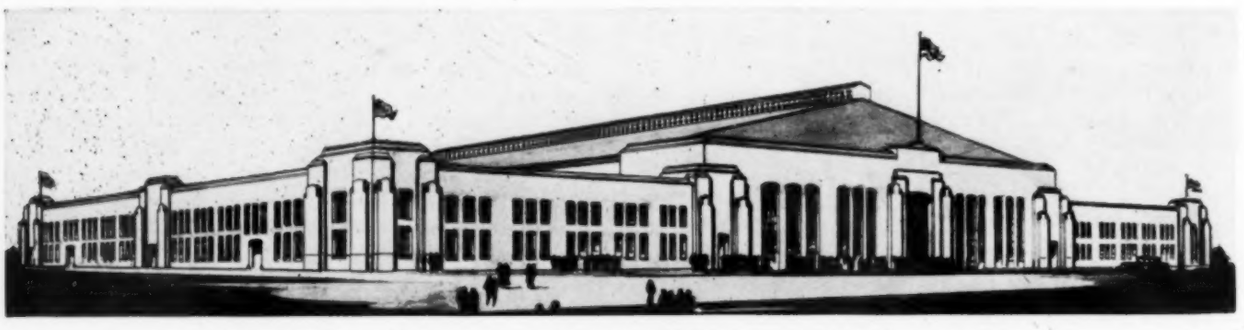
The growing national demand for modern heating and ventilating equipment indicates that the number of exhibitors and visitors at the Heating and Ventilating Exposition in Chicago will reach a new high. It is expected that the location in the mid-west will bring forth many new participants. The great Amphitheatre is a magnificent new structure, says Charles F. Roth, of the International Exposition Co., New York, N. Y., who will again be responsible for details of management.

The wide interest in new building construction and the increasing activity in the modernization of existing buildings gives promise that this Exposition will be one of the most important events in the fields of heating, ventilating, and air conditioning. Because it will be possible for exhibitors to operate their machinery and demonstrate the effective and economical methods of heating with coal, oil, and gas, and have fans, unit heaters, air conditioning, and refrigerating equipment in operation, the show will have added interest to those who attend.

The International Amphitheatre is new and is easily accessible by elevated, surface lines, or motor bus and special arrangements have been made for frequent service direct from hotel headquarters to the exposition building. For those desiring to drive to the Amphitheatre, a large private parking space is available without charge and accommodations for 10,000 cars are provided.

The Fourth International Heating and Ventilating Exposition will be held at the same time as the 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers and the Council has announced the Exposition Advisory Committee with the following membership: John Howatt, *Chairman;* J. M. Frank, F. H. Gaylord, R. E. Hattis, A. B. Martin, E. M. Mittendorff, and J. R. Vernon.

Invitations to participate in the Exposition have been extended to a number of organizations, whose interests are allied to heating, ventilating and air conditioning, and representatives of the following associations have been invited to serve on the committee: *American Gas Association, American Institute of Architects, American Oil Burner Association, American Society of Refrigerating Engineers, Concealed Heater Manufacturers Association, Heating, Piping & Air Conditioning Contractors National Association, Indirect Water Heater Manufacturers Association, Industrial Unit Heater Association, Institute of Boiler and Radiator Manufacturers, National Association of Air Filter Manufacturers, National Association of Fan Manufacturers, National Warm Air Heating & Air Conditioning Association, Steam Heating Equipment Manufacturers Association, Heat Exchange Institute, Steel Heating Boiler Institute, Stoker Manufacturers Association, Unit Air Conditioner Manufacturers Association, Vacuum and Condensation Pump Group Manufacturers,* and *Warm Air Register Manufacturers Institute.*



International Amphitheatre, Scene of the Fourth Heating and Ventilating Exposition

PRESIDENT'S MESSAGE TO MEMBERS

During March your President visited the Kansas City, Minnesota, Pacific Northwest, St. Louis, Southern California and the Wisconsin Chapters and in addition met the members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS residing in and about San Francisco and Oakland, California. Although the trip was of necessity a rather hurried one, I was able to learn by this direct contact of the sentiments and feelings of the members residing in the western half of the United States. I am happy to be able to say that the reception everywhere was most friendly and encouraging and since the Society must depend upon the activity and support of its Chapters, I feel indications are favorable for solidification and growth in the months ahead.

Little can ever be accomplished without enthusiasm. Nothing will destroy enthusiasm as quickly as will a lack of interest, so those Chapters that arrange meetings that are interesting to the members and to those in the community engaged in heating, ventilating, air conditioning and allied work are taking the first step necessary to expand the influence of the Society and to become important factors in their communities. The membership of a Chapter need not necessarily be large in order to be effective. Fifty active, enthusiastic members are of more influence and worth than five hundred listless, uninterested members. While there are Chapters with a larger membership than those visited by the President in March, the cheerful, cooperative, enthusiastic spirit shown by the members of these Chapters more than makes up for any lack of membership.

There are seventeen Chapters in the Society, each located in a center of population in the United States or Canada. Although some of these Chapters are thousands of miles apart in distance—all can be brought together into a closer union and closer to the National organization by the fostering of the spirit of friendliness—and helpful cooperation. The Chapters must appreciate that they are not separate independent units but are organized groups within the larger body of engineers that goes to make up the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Your Society officers can receive no more pleasing report than one that shows that each Chapter is doing well, that its meetings are interesting and well attended, and that it is making itself a factor in the affairs of its community.

April 1, 1935.

John Howatt
President,
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Local Chapter Reports

Illinois

March 11, 1935. The March meeting of the Illinois Chapter was held at the Hotel Sherman with an attendance of 112. Dinner was served at 6:45 p.m. in the Old Town Room and the meeting was called to order at 8:10 p.m. in the Crystal Room.

In the absence of Pres. R. E. Hattis and Vice-Pres. J. H. Milliken, the meeting was presided over by A. G. Sutcliffe. The minutes of the February meeting were read and approved.

The first order of business was the appointment of H. G. Kreissl, C. E. Crone, and H. P. Reid to act as tellers in the election of the Nominating Committee.

The acting secretary, J. J. Hayes, then read Pres. John Howatt's March letter to Chapters on the subject of Research, and two letters urging greater activity on the part of the Membership Committee. President Howatt expressed the desire that his home Chapter take the lead in this effort. A letter was read from A. V. Hutchinson, secretary, regarding regulations applying to membership qualifications.

A brief report followed, covering the Chapter's financial status.

Members were glad to learn from a recent postal card from their secretary, L. S. Ries, written from Florida, that he is rapidly regaining his health and anticipates returning to Chicago in the near future.

C. E. Price then introduced the speaker of the evening, Platte Overton, who spoke on Air Conditioning Existing Homes. Mr. Overton spoke of the public reaction to air conditioning and its sudden realization that homes were often dirty, hot, and humid. He predicted a wonderful future for the industry and seemed inclined to favor a combination of the direct and indirect system in order to avoid the so-called cold seventy. According to the speaker, in many modern homes with considerable glass area, it seemed necessary to provide a small convector for use after the fan stops to keep exposed glass and walls warm.

Mr. Overton outlined a number of his experiences in installing air conditioning in old homes and believed that the system should be designed around one room, its requirements determining the air temperature, velocity and volume, and the size of the boiler. In many cases, he found the ducts too small, requiring extremely high velocities; also coordination of different parts of systems appeared difficult, with dampers functioning not without trouble.

The talk was followed by numerous questions and much discussion by various members and guests. A rising vote of thanks was given to Mr. Overton for his paper.

The tellers then reported on the results of the election of the Nominating Committee. The following members who received the largest number of votes were declared elected as members of the Nominating Committee: John Howatt, J. J. Aeberly, J. J. Hayes, C. W. DeLand, H. R. Linn, R. E. Hattis, and J. H. Milliken.

The meeting adjourned at 9:45 p.m.

Michigan

March 11, 1935. Approximately 70 members and guests of the Michigan Chapter met for the regular monthly meeting held at the Wardell Hotel. This meeting was officially announced as the Annual University Night and the program was furnished by the University of Michigan and the University of Detroit.

The first speaker was Prof. Axel Marin of the University of Michigan. His discussion of Teaching Engineering brought out the recent changes which have been made in university curriculum, to broaden and supplement the technical training received in the engineering schools with business administration and law.

The latter part of the program was supplied by the University of Detroit. Prof. F. J. Linsenmeyer discussed the testing of vacuum pumps and stressed the often over-looked fact that the capacity of vacuum pumps as air eliminators is seriously reduced with increased temperature of return.

After his discussion Professor Linsenmeyer introduced in turn J. P. Schechter, J. J. McDonald, G. S. Rud, and B. B. Schuett, all of whom are students at the University of Detroit. They discussed effectively the relative merits of gravity warm air furnaces, steam heating, hot water heating, fan furnaces, and air conditioning in their application to residences. Their discussion was well received by the members of the Chapter.

The meeting was adjourned by Pres. G. D. Winans at 10:30 p.m. according to the report of the acting secretary, G. H. Tuttle.

Cleveland

February 26, 1935. The meeting was called to order at 8:30 p.m. when due to the severe weather only 15 members were present. The minutes of the January meeting were read and approved and the report of the treasurer was given. The secretary, E. J. Vermere, was instructed to send notices to all members who had not paid their dues, calling attention to the need for additional funds to carry on the work of the Chapter for the balance of the year.

Pres. M. F. Rather read a very complete report from W. E. Stark on the activities of the Council during their meetings in Buffalo in January.

H. E. Wetzell was introduced as the speaker of the evening and discussed the subject of Welding. Mr. Wetzell might be considered a pioneer in welding as applied to piping installations and he gave a very interesting account of the progress of pipe welding during the past ten or twelve years.

A detailed description of both the oxy-acetylene and electric arc welding processes was given. Mr. Wetzell spoke of the advantages and disadvantages of both processes and explained their adaptability to various types of work. A comparison was made between a welded pipe job and a screwed and flanged job. The speaker had a very complete display of standard welding fittings as well as some examples of poor workmanship found on installations where qualified workmen were not employed. The necessary tests which a welder must be able to pass today is a safeguard against this type of work. An interesting series of slides showed the history of pipe welding.

Before adjourning at 11:00 p.m., the Chapter voted this meeting the most interesting and instructive that has been held in a long time and recommends that other Chapters endeavor to hear Mr. Wetzell's address.

January 15, 1935. This meeting was held at Case School of Applied Science and was preceded by an inspection of the Mechanical Engineering Laboratory, where the 16 Student Members of the Society were hosts and in charge of testing equipment in operation. Of particular interest were 16 special test arrangements of heating, ventilating and air conditioning apparatus.

The business meeting was called to order at 8:30 p.m. with 35 members and guests present. Pres. M. F. Rather called attention to the Annual Meeting of the Society at Buffalo and T. A. Weager moved that President Rather be appointed delegate from the Cleveland Chapter. This nomination was seconded and carried.

Prof. C. A. McKeeman announced that Case School was starting an evening class in air conditioning and explained the subjects to be followed in both class and laboratory work.

Prof. G. L. Tuve introduced Dr. J. J. Nassau, head of the Department of Astronomy at Case School, as the speaker of the evening. Dr. Nassau chose the subject, Why Build Large Telescopes, and first explained how heating is allied with astronomy in determining surface and interior temperatures of the stars.

As a concrete example of the need of large telescopes, Dr. Nassau showed by means of slides, first the 10 or 15 stars in a certain area of the sky visible to the eye; next, increasing the number in the same area to several hundred by means of a good pair of field glasses; then, photographs of the same area through various sized telescopes until many thousands can be located by means of the 100 in. telescope at Mt. Wilson Observatory.

A very interesting and detailed description was given of the work already done in building the 200 in. reflecting telescope now under construction in Cleveland; also of the mechanism, which will weigh over 400 tons, required for moving it in order to keep it in focus.

A rising vote of thanks was extended to the speaker and the meeting adjourned at 10:45 p.m.

New York

March 18, 1935. A two-reel sound movie film on the subject of sound and noise prevention with an accompanying lecture by J. S. Parkinson, and a paper by E. D. Milener of the *American Gas Association* on Summer and Winter Air Conditioning with Gas were the features of the regular meeting of the New York Chapter held in the Building Trades Club. Pres. H. W. Fiedler presided.

The film showed graphically the difference between sounds of high frequency and those of low frequency and the difference in amplitude of sounds. It covered the fundamentals of sound in a non-technical way and showed a number of acoustical phenomena such as refraction, reverberation, echoes, etc.

Mr. Parkinson showed how noise is measured and used as an example a noise manufacturing machine from which noise issued through a tube, in which the speaker inserted a felt-lined tube to show how sound was absorbed by this material.

Mr. Parkinson told of the work of the Technical Advisory Committee on Sound in Relation to Heating and Ventilation of the Committee on Research of the Society in an attempt to classify all types of heating and ventilating equipment according to the type of sound emitted and the loudness of such sound; and in addition to set-up maximum standards for the amount of noise permissible in a particular type of room. Knowing these two things—the amount of noise a given machine will produce and the amount of noise that is permissible, the engineer will know at once how much noise prevention he must accomplish.

Mr. Milener briefly reviewed the accomplishments of the gas industry in automatic heating and described the work which the Industrial Gas Section of the *A.G.A.* has done in developing a summer conditioning unit, which will absorb moisture for dehumidifying. In the speaker's opinion, more installations of summer air conditioning can be satisfactorily supplied by dehumidifying the air than by mechanical cooling with dehumidifying as a by-product. The speaker then described a gas unit commercially available for accomplishing dehumidifying, with silica gel as the adsorbent.

R. H. Carpenter, A.S.H.V.E. Council member, reported on the subject of the method of electing the A.S.H.V.E. Nominating Committee. Several different methods have been proposed as a change from the present method in order to give all members fair representation. Among these are retaining the present method of allowing no member of the Nominating Committee to serve for two consecutive years. An alternate method is to let the Council select the Nominating Committee; while a third proposal is to have the presidents of the Chapters select the Committee. Considerable discussion followed and President Fiedler appointed a committee of five, consisting of W. H. Driscoll, Arthur Ritter, H. B. Hedges, W. W. Timmis, and R. H. Carpenter, to report back to the Chapter at the next meeting.

Nominations for Chapter officers for the coming season were made by the Nominating Committee as follows:

President—W. W. Timmis.

Vice-President—G. E. Olsen.

Secretary—T. W. Reynolds.

Treasurer—W. M. Heebner.

Board of Governors—H. W. Fiedler, A. F. Hinrichsen, W. E. Heibel.

Ontario

March 4, 1935. The regular meeting was held at the Royal York Hotel, Toronto, and 64 members and guests were present. Dinner was served at 6:30 p.m. and Vice-Pres. M. W. Shears acted as chairman of the meeting in the absence of Pres. W. R. Blackhall.

Following dinner, Secy. H. R. Roth, read the minutes of the preceding meeting, which were accepted as read. The secretary reported that the Board of Governors had made selections of

chairmen of the various committees for the Semi-Annual Meeting, as follows: E. H. Gurney, *General Chairman;* L. L. Anthes, *Vice-Chairman;* Thomas McDonald, *Banquet Committee;* H. H. Angus, *Reception Committee;* M. F. Thomas, *Finance Committee;* E. R. Gauley, *Publicity Committee;* J. S. Paterson, *Registration Committee;* W. P. Boddington, *Entertainment Committee;* H. B. Jenney and Mrs. E. H. Gurney, *Ladies Committee;* M. W. Shears, *Transportation Committee;* and H. R. Roth, *Secretary.*

The secretary read the report of the Nominating Committee for the selection of candidates for office for the year 1935-36:

President—M. W. Shears.
Vice-President—Thomas McDonald.
Secretary-Treasurer—H. R. Roth.
Board of Governors—H. D. Henion, G. A. Playfair, H. B. Jenney.

It was moved, seconded, and carried, that the Ontario Chapter loan $100.00 to the Finance Committee for the 1935 meeting of the Society to meet committee expenditures; this money to be repaid to the treasurer of the Chapter from the funds provided for the meeting.

A letter to the Chapter on Research from Pres. John Howatt of the A.S.H.V.E. was read by Mr. Shears.

O. L. Maddux introduced the principal speaker of the evening, E. D. Milener, *American Gas Association,* who addressed the gathering on An Hour's Discussion of Gas Heating and Year Around Air Conditioning. Mr. Milener displayed a number of slides and showed one motion picture reel, entitled, Gas Heating, which was of interest to the members.

Following Mr. Milener's splendid address, a vote of thanks was extended by Mr. Watson.

Pacific Northwest

February 26, 1935. Pres. A. L. Pollard presided at the meeting of the Pacific Northwest Chapter, when 35 guests attended.

Among the guests was N. D. Adams, Rochester, Minn., who gave an instructive talk on the mechanical equipment of the Mayo Clinic. Mr. Adams' talk was accompanied with slides and was greatly appreciated and enjoyed by the Chapter.

Prof. E. O. Eastwood gave some facts regarding the power plant and central heating system of the University of Washington campus. He then introduced G. H. Krueger, who at the present time is completing plans and specifications for an addition to the existing plant at the University.

W. E. Beggs reported on progress being made with the State Legislature in an effort to have enacted a professional engineer's registration law in the State of Washington.

E. L. Weber described and demonstrated an electric differential thermometer, which he has designed and constructed to balance hot water heating systems. The instrument measures in degrees the temperature drop in any individual radiator and its drop can be varied by adjustment of a non-reflux equalizing elbow, which Mr. Weber has patented.

According to the report of Secy. S. D. Peterson, the meeting adjourned at 10:00 p.m.

January 22, 1935. The January meeting of the Pacific Northwest Chapter was held at the Hungerford Hotel with 18 members and guests present, and Pres. A. L. Pollard presiding.

M. J. Hauan was selected as representative and M. N. Musgrave was designated as alternate for the Chapter on the A.S.H.V.E. Nominating Committee.

One of the speakers of the evening was Mr. Musgrave, who gave a very interesting talk on electrical temperature control systems.

The guest speaker was G. A. Zanboni, who presented an illustrated talk on the production and manufacture of lubricating oils.

The meeting adjourned at 9:30 p. m.

Western Michigan

February 11, 1935. Dinner was served at 7:00 p.m. in the Union Parlors to 14 members and 19 guests of the Western Michigan Chapter who met at Michigan State College, East Lansing, Mich. Pres. S. H. Downs called the meeting to order and minutes were read by Secy. W. G. Schlichting and approved.

President Downs then read a communication from John Howatt, Chicago, Ill., president of the A.S.H.V.E. Acting upon a motion presented by J. H. Van Alsburg, the secretary was directed to write President Howatt to extend the greetings and pledge the cooperation of members of the Western Michigan Chapter.

Reports on the Annual Meeting in Buffalo were given by Mr. Van Alsburg and President Downs.

The Chapter was fortunate in having as their distinguished guest, L. P. Breckenridge, North Ferrisburg, Vt., professor emeritus of Mechanical Engineering, Yale University. In his talk, Professor Breckenridge traced briefly the progress which has been made in the science of heating and ventilation during the last half century.

Professor Breckenridge introduced Dean H. B. Dirks, who discussed the paper, Summer Cooling in the Research Residence for the Summer of 1934, presented at the 41st Annual Meeting of the Society.

The next speaker was Prof. L. G. Miller, who discussed the paper, The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, which had been given by Dr. C.-E. A. Winslow at the Buffalo Annual Meeting. Professor Miller also described and demonstrated a similar instrument, the Kata thermometer.

This very interesting meeting did not adjourn until 11:00 p.m.

January 14, 1935. The meeting was held in the English Room of the Rowe Hotel, Grand Rapids, Mich., when dinner was served to 9 members and 4 guests. Pres. S. H. Downs called the meeting to order and minutes of the previous meeting were read by Secy. W. G. Schlichting and approved.

Members were urged to attend the Society's Annual Meeting at Buffalo, N. Y., and a letter was read from A. V. Hutchinson, secretary, in regard to this event.

N. S. Yost, Howell, Mich., was then introduced and outlined the various types of alternating current motors and their control, describing fully their characteristics and applications. Of particular interest was his classification of motors in regard to quietness and his outline of the factors contributing to noise.

After a general discussion, the meeting adjourned at 10:00 p.m.

Wisconsin

February 18, 1935. The meeting of the Wisconsin Chapter was held in the Mirror Room of the Hotel Pfister and, following dinner, was called to order by Pres. Ernest Szekely. Due to the absence of the secretary, G. E. Hochstein, the reading of the minutes of the previous meeting was dispensed with.

President Szekely stated that Pres. John Howatt of the A.S.H.V.E. would send a monthly letter to the Chapter advising of the national activities of the Society. The February letter was read and gave a few sidelights on the Annual Meeting at Buffalo. It was suggested that President Howatt's letter be acknowledged by the secretary and President Szekely announced that Mr. Howatt would visit the Wisconsin Chapter on March 19.

The first speaker of the evening was L. A. Burch, who presented an interesting summary of Chapter 1, Fundamentals of Heating and Air Conditioning, of THE A.S.H.V.E. GUIDE 1935.

The meeting was then turned over to Vice-Pres. C. H. Randolph who introduced the guest speaker, Mr. Rogers.

A talk on the development of water supplies from deep wells was presented by Mr. Rogers, and was of special interest to Chapter members from the standpoint of providing cold water supplies for use in connection with air conditioning. Mr. Rogers supplemented his address with an interesting movie reel, showing the development of a well in drift formation.

The meeting was closed by Vice-President Randolph at 9:55.

Mrs. James A. Donnelly Dies

Friends of Mrs. James A. Donnelly will be shocked to learn of her death at Orlando, Florida, on January 29, 1935, for as Miss Helen R. Innis and later as Mrs. Donnelly she was well known to Society members and was a familiar figure at A.S.H.V.E. meetings.

Mrs. Donnelly was born on June 3, 1892 in Brooklyn, N. Y., and was one of the first women to become interested in heating and ventilating engineering. She attended the New York School of Heating and Ventilating and also took a steam power course at Pratt Institute, Brooklyn. She first affiliated with the Society as a Junior Member in 1918 and in 1921 was advanced to the grade of Member.

For a number of years Mrs. Donnelly was associated with Positive Differential System Co. as secretary and treasurer and later with Donnelly Systems Corp. In addition to her executive duties, she supervised the laying out of heating systems for residence and industrial plants where vacuum or vapor systems were to be installed.

During recent years, Mr. and Mrs. Donnelly have made their home at Largent Springs Manor, Largent, W. Va., and have spent several winters in Florida.

The Officers and Council join in expressing their sincere sympathy to Mr. Donnelly, their fellow member, and are keenly aware of the loss which has been sustained in the death of Helen Innis Donnelly.

Hugo Junkers

Hugo Junkers, who became a Member of the A.S.H.V.E. in 1925, died on his 76th birthday, February 3, 1935, at Munich, Germany. He was born in the Rhenish city of Rheydt and received an extended schooling and training. After his gymnasium education he studied mechanical engineering at the polytechnical schools in Berlin, Karlsruhe, and Aix-la-Chapelle. He passed the government examination as an engineer's assistant when he was 25 years old.

After several years of private experimental work, he began producing inventions in heat engines and apparatus, which was interesting to him but not very lucrative. In 1895 he started the firm of Junkers & Co. in Dessau to make heating apparatus, specializing in portable gas stoves. His business ventures were not always profitable, however, and from 1897 until 1910 he was professor of heat engines and the technic of heat at the Polytechnic School in Aix-la-Chapelle, Germany.

With his combined incomes he was able to delve deeply into the then unfathomed secrets of oil-driven motors and heavier-than-air machines. The first all-metal plane to fly successfully was the result of his research. Later he specialized in gas motors almost exclusively at Dessau and developed an all-metal monoplane of an underslung type. Planes of his manufacture have held various world flying records, one of the most notable being the first flight of a heavier-than-air machine from Europe to America made by the Bremen in April 1928.

Mr. Junkers wrote voluminously on his findings and he was a recognized authority on airplane construction and highly regarded in the field of heat technics.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 60 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1935, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	*REFERENCES*	
	Proposers	*Seconders*
AYERS, E. H., Supt., Healy Plbg. & Htg. Co., St. Paul, Minn.	Albert Buenger E. A. Jones	C. E. Gausman D. W. Hickey
BARTON, D. H., Student, Agric. & Mech. College of Texas, College Station, Tex.	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
BENSEN, C. L., Engr., McQuay Inc., Minneapolis, Minn.	A. B. Algren F. B. Rowley	R. E. Backstrom Albert Buenger
BLACKHALL, L. C., Engr., Gurney Foundry Co., Ltd., Toronto, Ont.	E. H. Gurney W. R. Blackhall	G. E. Cole J. W. O'Neill
BLESSED, W. A., Mech. Engr., Mueller Brass Co., Port Huron, Mich. *(Reinstatement and Advancement).*	G. D. Winans W. G. Boales	J. H. Walker E. E. Dubry
BURKS, R. H., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
CAWBY, E. L., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne J. N. Hirst	Margaret Ingels L. L. Lewis
CHRISTOPHERSEN, A. E., Engr.-Custodian, Board of Education, Chicago, Ill.	E. V. Hill J. P. Fleming	E. B. Ericsson J. J. Finan
COCHRAN, W. B., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*

CORRIGAN, J. A., Engr., Corrigan Co., St. Louis, Mo. *(Advancement.)*	C. M. Humphreys T. F. Rockwell	L. W. Moon E. A. White
COTTER, R. P., Student, Agric. & Mech. College of Texas, College Station, Texas	C. W. Crawford *(A.S.M.E.)* V. M. Faires *(A.S.M.E.)*	R. M. Wingren *(A.S.M.E.)* J. A. Trail *(A.S.M.E.)*
DE BOOS, F. A., Dist. Mgr., B. F. Sturtevant Co., Detroit, Mich.	J. S. Kilner G. H. Cummins	A. P. Darlington H. E. Paetz
EARLEY, T. J., Sales Engr., Jennison Co., Fitchburg, Mass.	R. T. Kern J. W. Brinton	H. B. Wiegner W. A. McPherson
ECKERT, E. K., Htg. & Air Cond. Engr., Holland Furnace Co., Knoxville, Tenn.	V. W. Cherven H. Westrate *(Non-Member)*	J. N. Sawyer Stanley Curtis *(Non-Member)*
EDWARDS, H. B., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne J. N. Hirst *(Non-Member)*	Margaret Ingels L. L. Lewis
FEELY, F. J., Mgr. of Sales, The Taylor Supply Co., Detroit, Mich. *(Advancement.)*	W. G. Boales G. D. Winans	A. C. Wallich R. K. Milward
FITZGERALD, W. E., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
GRAHAM, E. W., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne J. N. Hirst	Margaret Ingels L. L. Lewis
GRANSTON, R. O., Engr. and Estimator, University Plbg. & Htg. Co., Seattle, Wash. *(Advancement).*	W. W. Cox S. D. Peterson	R. O. Wesley *(Non-Member)* H. A. Cook
HAMIG, L. L., Sales Engr., Chrysler Air Cond. Div., Weber Implement & Auto Co., St. Louis, Mo.	A. L. Walters F. J. McMorran *(Non-Member)*	E. A. White L. W. Moon
HARTON, A. J., Sales Engr., St. Jos. Ry. Lt. Ht. & P. Co., St. Joseph, Mo.	G. E. Olson L. R. Chase	C. C. Cheyney T. L. Dawson
HAYMAN, A. E., Jr., Journeyman, Albert E. Hayman, Wilmington, Dela. *(Advancement.)*	C. M. Humphreys F. C. McIntosh	T. F. Rockwell W. H. Osterle
HENSZEY, W. P., Jr. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne L. L. Lewis	D. E. French J. N. Hirst
HICKS, H. K., Student, Swarthmore College, Swarthmore, Penna.	W. W. Timmis M. E. Barnard *(Non-Member)*	C. G. Thatcher *(A.S.M.E.)* H. M. Jenkins *(Non-Member)*
HILDRETH, L. W., N. Y. Mgr., Anthracite Institute, New York, N. Y.	J. M. Traugott G. L. Larson	D. W. Nelson G. C. Wilson *(Non-Member)*
HOEHL, E. R., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne J. N. Hirst	Margaret Ingels L. L. Lewis
HOWLETT, I. G., Consulting Engr., Howlett-Tauson Co., Oklahoma City, Okla. *(Advancement.)*	R. G. Dolan S. L. Rolland	E. W. Gray E. F. Dawson
ILLIG, W. R., Owner, W. R. Illig Plbg. & Htg., Fitchburg, Mass. *(Reinstatement.)*	C. T. Flint F. R. Ellis	I. J. Kelley W. T. Jones
JONES, A. P., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
JORDAN, R. C., Charge of Air Cond. Dept., Central Supply Co., Minneapolis, Minn. *(Advancement.)*	F. B. Rowley A. B. Algren	G. C. Morgan G. A. Dahlstrom
KAUFMAN, C. W., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne L. L. Lewis	D. E. French J. N. Hirst
KELLY, W. C., Field Engr., Iron Fireman Mfg. Co., Ltd., Toronto, Ont., Can.	H. S. Moore T. McDonald	W. R. Blackhall A. S. Leitch
LEVELTHAN, BERNARD, Student, Polytech. Inst. of Brooklyn, Brooklyn, N. Y.	John Everetts, Jr. Gerrado Immediato *(Non-Member)*	E. F. Church, Jr. *(Non-Member)* Moris Kurzweil *(Non-Member)*
MCCRIMMON, A. M., Asst. Secy. & Controller, Hydro-Elec. Power Commission, Toronto, Ont.	E. H. Gurney A. S. Leitch	S. L. Fear *(A.S.M.E.)* J. S. Wood
MCDOWELL, B. W., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne L. L. Lewis	D. E. French J. N. Hirst
MCGAUGHEY, J. E., Jr., Air Cond. Engr., Carrier Engrg. Corp. New York, N. Y.	R. E. Cherne L. L. Lewis	D. E. French J. N. Hirst
MIDDLETON, H. A., Proprietor, Middleton Elec. Co., Sedalia, Mo.	L. R. Chase E. K. Campbell	H. L. Stevens H. M. Rudio
MONIER, K. A. J., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
NOORD, D. F., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys A. McGonagle	G. S. McEllroy *(Non-Member)* F. C. McIntosh
PAUL, L. O., Air Cond. Engr., Carrier Engrg. Corp., Chicago, Ill.	R. E. Cherne J. N. Hirst	Margaret Ingels L. L. Lewis
PRIESTER, G. B., Air Cond. Engr., Carrier Engrg. Corp., New York, N. Y. *(Advancement).*	R. E. Cherne J. N. Hirst	Margaret Ingels L. L. Lewis
RODGERS, W. C., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*

Roos, E. B. J., Air Cond. Engr., Carrier Engrg. Corp., Newark, N. J.	R. E. Cherne J. N. Hirst	Margaret Ingels L. L. Lewis
Rosenburg, W. E., Assistant to J. C. Rosenburg, Locust Valley, L. I., N. Y.	A. A. Adler M. C. Giannini	W. W. Timmis W. R. Zuhlke
Runkel, Charles, Pres., Acme Htg. & Vtg. Co., Chicago, Ill.	J. W. Mathis John Howatt	J. J. Finan E. Mathis
Schucany, O. W., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
Schwantes, A. R., Engr., The Waterman-Waterbury Co., Minneapolis, Minn.	F. W. Legler F. G. Sedgwick *(Non-Member)*	H. G. Cross *(Non-Member)* F. B. Rowley
Seelert, E. H., Factory Mgr., McQuay, Inc., Minneapolis, Minn.	A. B. Algren F. B. Rowley	R. E. Backstrom Albert Buenger
Sluss, A. H., Prof., Mech. Engrg., University of Kansas, Lawrence, Kan.	E. K. Campbell E. K. Campbell, Jr.	L. A. Stephenson Carl Clegg
Spencer, J. B., Pres. & Chief Engr., Spencer Cooling & Air Conditioning Co., Minneapolis, Minn.	R. W. Otto G. C. Morgan	Albert Buenger A. B. Algren
Stringfellow, J. C., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
Tauson, P. O., Consulting Engr., Howlett-Tauson Co., Oklahoma City, Okla. *(Advancement.)*	R. G. Dolan S. L. Rolland	E. W. Gray E. F. Dawson
Teal, E. T., Student, Agric. & Mech. College of Texas, College Station, Texas	R. M. Wingren *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*	V. M. Faires *(A.S.M.E.)* J. A. Trail *(A.S.M.E.)*
Tiller, Louin, Design Engr., Okla. Gas & Elec. Co., Oklahoma City, Okla. *(Advancement).*	S. L. Rolland E. W. Gray	R. G. Dolan B. R. Miller
Tucker, L. A., Chief Engr., J. J. Pocock, Inc., Philadelphia, Pa.	L. C. Davidson J. H. Hucker	C. B. Eastman R. F. Hunger
Warren, C. W., Student, Agric. & Mech. College of Texas, College Station, Tex.	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
Weatherby, E. P., Jr., Student, Agric. & Mech. College of Texas, College Station, Tex.	R. M. Wingren *(A.S.M.E.)* V. M. Faires *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
Wilson, J. W., Student, Agric. & Mech. College of Texas, College Station, Texas	V. M. Faires *(A.S.M.E.)* R. M. Wingren *(A.S.M.E.)*	J. A. Trail *(A.S.M.E.)* C. W. Crawford *(A.S.M.E.)*
Wright, C. E., Part-time Instructor, Carnegie Inst. of Tech., Pittsburgh, Pa. *(Advancement.)*	C. M. Humphreys F. C. Houghten	T. F. Rockwell W. H. Osterle
Zangrilli, A. J., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	T. F. Rockwell C. M. Humphreys	John Richmond H. F. Huettner

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

Bloom, Louis, Co-Partner, Bloom Htg. Co., Brooklyn, N. Y.

Danielson, W. A., Major, U. S. Army (Quartermaster Corps), Washington, D. C.

Graham, C. H., Sales & Engrg., Lennox Furnace Co., Syracuse, N. Y.

Hausman, L. M., Pres. & Gen. Mgr., L. M. Hausman & Co., Manila, Philippine Is.

Herske, A. R., American Radiator Co., New York, N. Y. *(Reinstatement.)*

Marin, Axel, Assoc. Prof. Mech. Engrg., Univ. of Mich., Ann Arbor, Mich.

Trostel, O. A., Engr., Kern Engrg. Co., Inc., Milwaukee, Wis.

Wardell, Arthur, Partner, Thomas & Wardell, Toronto, Ont., Can.

Wolfert, Edward R., Section Engr., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

ASSOCIATES

Baker, W. H., Jr., Sales Engr., Crane Co., Chicago, Ill.

Cook, H. R., Mgr., American Foundry & Furnace Co., Milwaukee, Wis. *(Reinstatement.)*

Dawson, G. S., Mdse. Sales Engr., B. C. Elec. Rly. Co., Ltd., Vancouver, B. C.

Dever, H. F., Sales Engr., Minneapolis-Honeywell Reg. Co., Minneapolis, Minn.

Gamble, C. B., Crawford, Inc., New Orleans, La.

Hochstein, G. E., Sales Engr., Heil Co., Milwaukee, Wis.

Krez, Leonard, Secy., Paul J. Krez Co., Chicago, Ill.

Scudder, Barrett, Vice Pres., J. P. Marsh Corp., Chicago, Ill.

Swift, P. F., Mgr., The Carl F. Scheffer Co., Dayton, Ohio.

JUNIORS

Groves, S. A., Junior Salesman, American Radiator Co., New York, N. Y.

Hanson, L. P., Engr., U. S. Air Cond. Corp., Minneapolis, Minn. *(Advancement.)*

Meakin, J. B., Grad. Student, Mass. Inst. of Tech., Boston, Mass.

Moore, C. H., Efficiency Mgr., Congress Hotel Co., Lake Forest, Ill.

Stitt, A. B., Plbg. & Htg. Engr., Sears, Roebuck & Co., Stamford, Conn. *(Advancement.)*

STUDENTS

Baur, J. W., Student, University of Ill., Urbana, Ill.

Boehmer, A. P., Student, University of Ill., Urbana, Ill.

Compton, W. E., Grad. Student, University of Ill., Urbana, Ill.

Farbman, S. M., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

Frisse, J. L., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

Goodman, Daniel, Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

Greiner, G. E., Jr., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

Moeller, Robert, Student, Case School of Applied Science, Cleveland, Ohio.

Neis, W. A., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

Schnitzer, Sidney, Student, University of Ill., Urbana, Ill.

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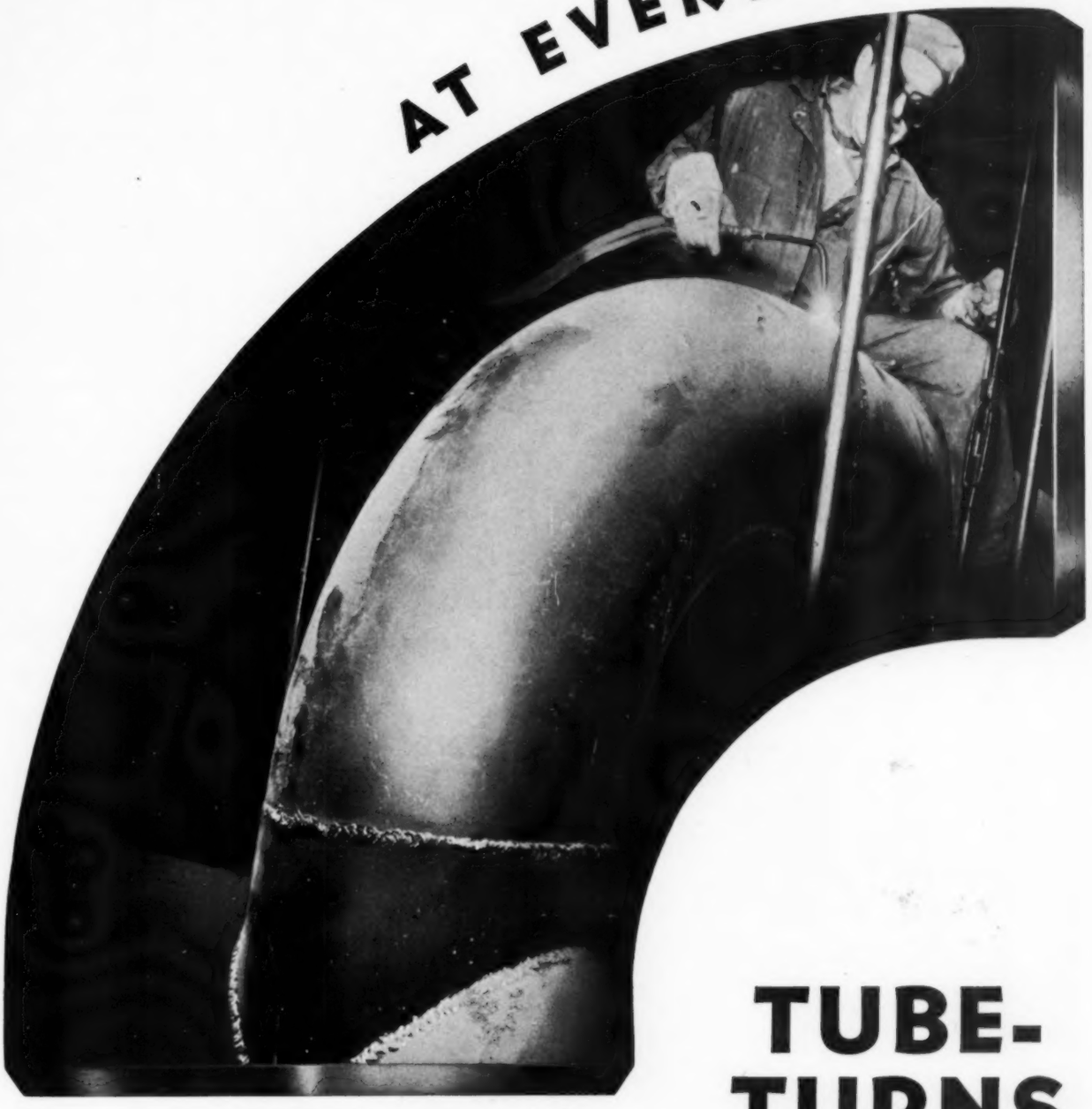

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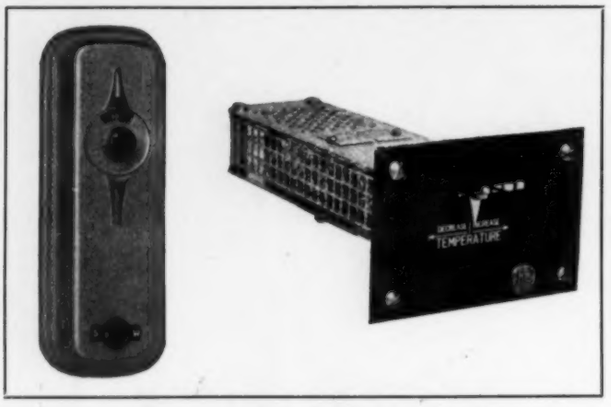
No. 300—An advanced range of thermostats, designed especially for air conditioning requirements are sensitive to temperature changes as low as plus or minus ½ F, states the maker,



and have special features that eliminate chatter. Summer-winter manual switches are built into the instrument case, and interwired so that all types of heating and cooling equipment can be operated off a single instrument during both summer and winter. Switches are either of the ordinary toggle type or can be furnished for key operation for use in public buildings.

The summer-winter switch reverses action of the electrical contacting within the instrument so that heating equipment is started on falling temperature in the winter, cooling equipment is started on rising temperature in the summer. Cases are of handsome design finished on the "Alumilite" or anodized process to be proof against corrosion and scratches.

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The compressor is a two-cylinder freon unit with 1¾ in. bore and 1¾ in. stroke driven through V-belts by a ¾ hp motor. The 9 in. radial type fan circulates about 270 cfm, of which approximately 25 per cent may be outside air. Hand damper control for outdoor air is incorporated in the cabinet.

Cleanable air filters are provided on both outdoor air intake and recirculated air intake.

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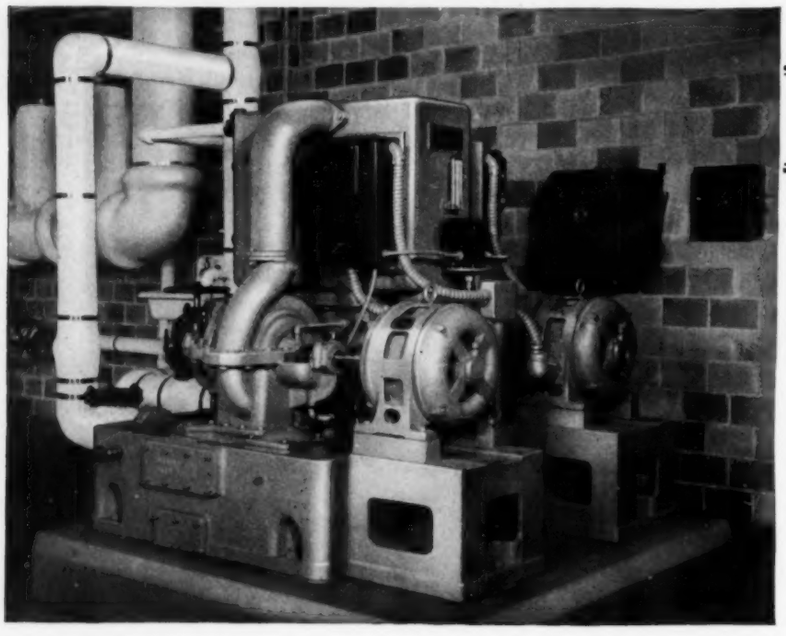

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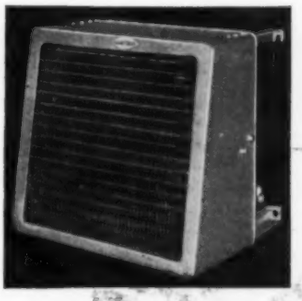

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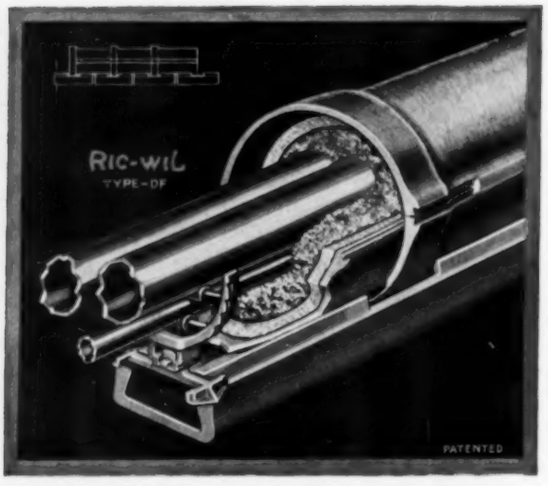

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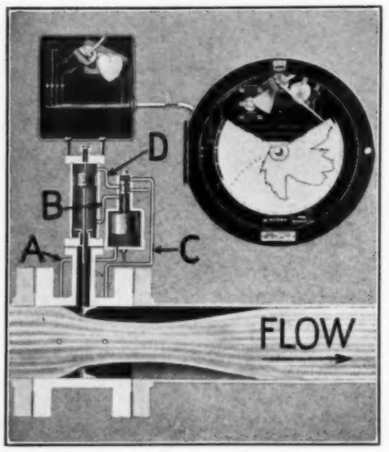
Flow Meter Uses Varying Orifice

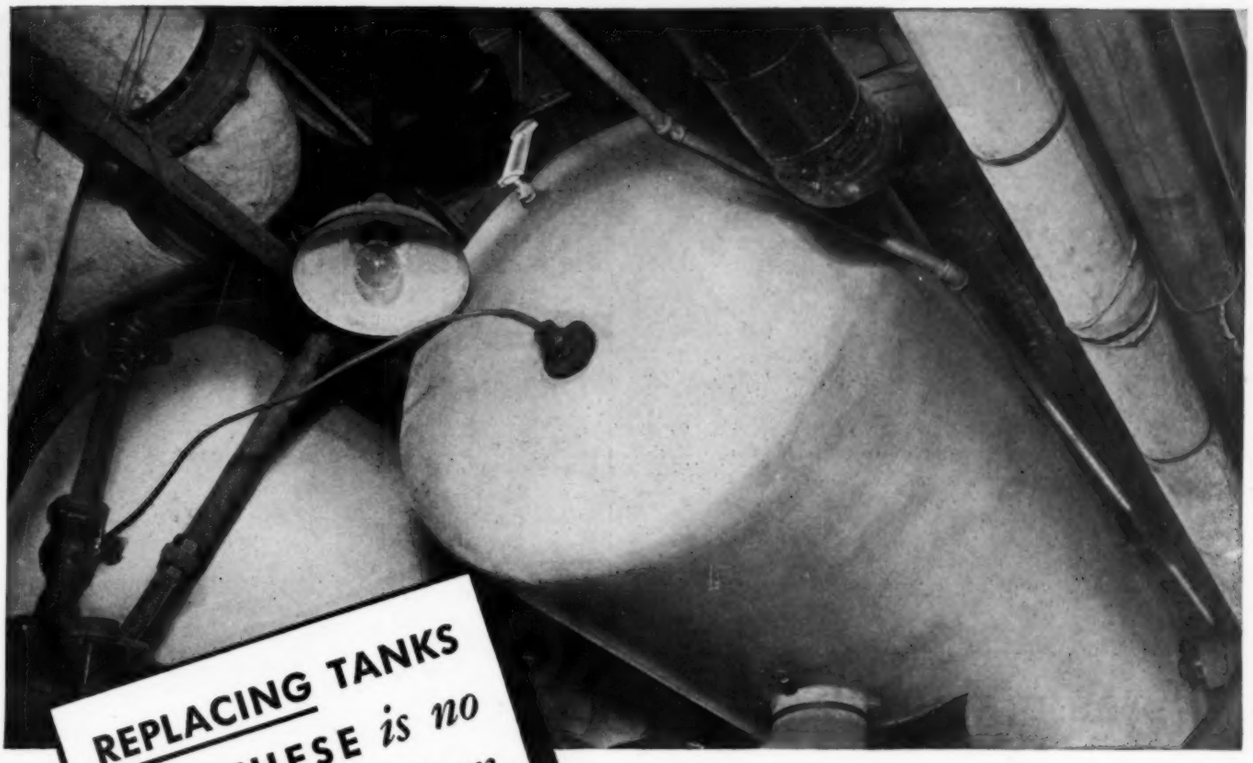
No. 302—A new type of flow meter utilizes a varying orifice; as the flow of steam, water, oil, or other fluid varies, the orifice opens and closes. Thus a constant pressure drop is maintained by changing the orifice opening and flow is determined by measuring electrically the amount of orifice opening.



It is claimed that the varying orifice principle makes it possible to measure accurately low rates of flow down to 2½ per cent of the meter capacity, and that pulsating flows can be more accurately measured.

The complete meter consists of two parts, the actuator and the meter unit. The former is in the pipe line and is a flanged cylindrical section in the center of which is mounted a set of plates, forming a variable orifice. Upstream pressure enters pilot through tube "A" (see cut) beneath piston and causes pilot piston to rise, permitting upstream pressure to enter main cylinder through tube "B" beneath main piston; this raises main piston and opens orifice. Orifice will continue to open until downstream pressure reaches desired point, at which time pilot piston travels downward and shuts off pressure under main piston, orifice opening being held in fixed position.

If flow increases, pilot piston again rises, admitting pressure again below main piston and causes orifice to open further. If flow in main pipe line drops off, downstream pressure, which is transmitted to top of pilot piston through tube "C," will cause pilot piston to travel downward, admitting pressure to top of



REPLACING TANKS LIKE THESE *is no "pin money" item*

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DURONZE

FOR PERMANENCE

CORRODIBLE tanks may have a short life . . . but it's not a merry one, especially for the owner. Figure it how you will—it pays to use Duronze at the very start, and *avoid* replacement costs.

Duronze is a silicon bronze (high copper alloy) which is about twice as strong as copper and as tough as steel. It can be fabricated into tanks by exactly the same tools and methods as tanks made from steel.

Duronze is supplied to tank manufacturers in the form of large sheets or plates which are formed, then readily welded by either acetylene gas or electric arc methods.

Specify Duronze for storage tanks, range boilers, automatic heaters, ducts and flues—in fact, for any purpose requiring the strength of steel and the corrosion-resistance of copper. Our Engineering Department will be glad to furnish you with full information on request. Write today.

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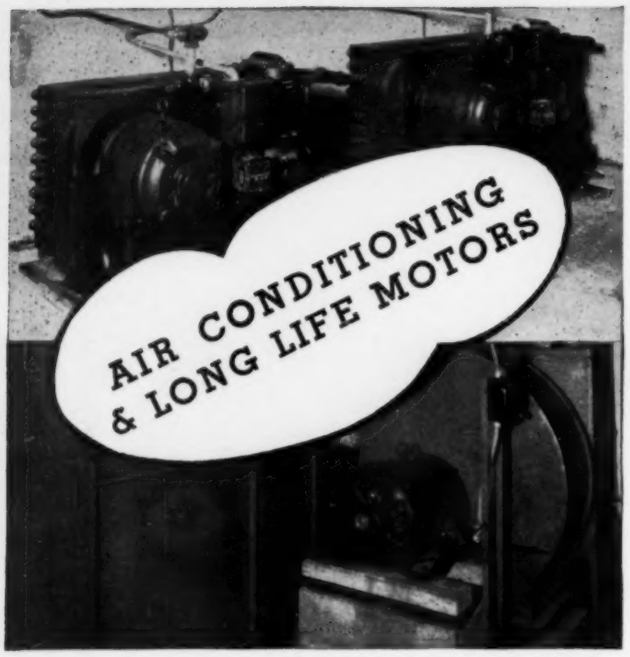

REPULSION START INDUCTION BRUSH LIFTING SINGLE PHASE MOTORS

High starting torque — low starting current — dependable — efficient! Look for these features in motors for your Air Conditioning Equipment — you'll find them all in Century Single Phase Brush Lifting Type Motors, proved by more than 31 years' successful service in automatic and remote control installation under all kinds of surrounding climates and severe operating conditions...Sizes 1/8 to 40 Horse Power.

Have you an Air Conditioning design or installation problem? Century Engineers will help you — Century Motors will satisfy every requirement of this modern development.

Century Motors are built in sizes 1/250 to 600 Horse Power.

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Century

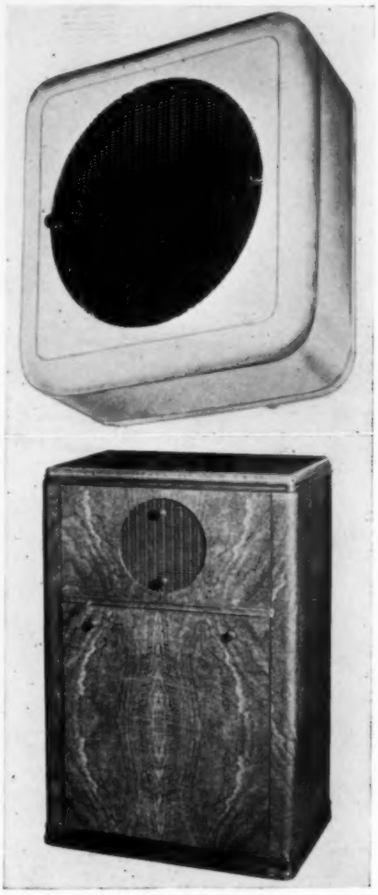
SINGLE PHASE, POLY-PHASE, DIRECT CURRENT
1/250 to 600 H. P.
MOTORS

main piston through tube "D," and opening a drip line venting main cylinder at bottom of main piston. This causes the orifice to close until required pressure drop again is established. Electrical impulses from the actuator may be transmitted by a two wire circuit to any remote location. Several combinations of meters may be operated by one actuator and the meters may be located in various parts of the plant.—Elgin Softener Corp., Elgin, Ill.

New Ceiling Coolers and Spot Cooler

No. 303—A new line of ceiling type unit coolers is composed of four sizes ranging from 1 to 4 tons cooling capacity.

Directional air control, up or down, to the right or to the left, is afforded by the circular adjustable grille, permits concentration of the cooled air in any direction if desirable. Front panel of the unit is hinged and is easily opened revealing the expansion valve and coil. All connections are made in the back where the drain and supply and return refrigerant lines are grouped at the base of the fan support. Fan unit is protected by a two section removable guard.



The units are intended for use singly or in multiple with a remotely located compressor; freon, methyl chloride or cold water can be used as refrigerants.

Also announced recently is a new "Spot Kooler" equipped with a water cooled compressor located in the base of the grained metal cabinet. The fan and coil are in the top of cabinet. The circular adjustable grille provides concentration of the cooled air to the right or to the left, and up or down. Connections required are to the electric light circuit and water supply and return lines.

The cooler is 42¾ in. high, 29¼ in. wide, and 18½ in. deep. —Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, Ill.

Hot Water Circulating Pump Improved

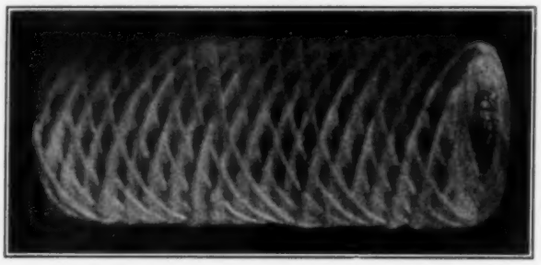
No. 304—Development of a new hot water circulating pump for use with the "Force-Flo" one pipe system of hot water heating and other hot water applications has just been announced. The pump employs a new type water seal which replaces the stuffing box formerly used and which is capable of operating against high pressures without leakage, states the maker. Another new feature is the forged impeller which combined with reduced friction made possible by the seal gives greater capacities at higher heads than formerly obtained. It is made in five sizes from ¾ to 3 in., the largest size delivering 100 gpm at 5½ ft of friction head.

This company also has announced the development of new

control valves for checking hot water circulation, indirect heaters, and other specialties which are combined into a complete system of space and domestic water heating for summer-winter operation from the same boiler.—The Stevens-Root Co., 51 E. Grand Ave., Chicago, Ill.

Honeycomb Tube for Filtering Liquids

No. 305—A filter with honeycomb filter tube for filtration of oils, milk, water, fruit juices, alcoholic liquors, and other liquids has an element comprising a tubular body of bleached cotton yarn which is 8 in. long and 2½-2¾ in. in diameter. The yarn is wound by a special process to form a honeycomb of deep



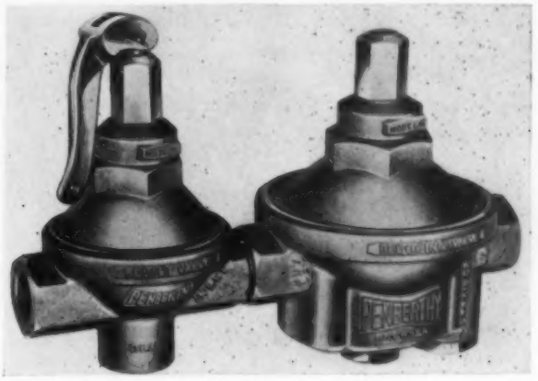
tapered tunnels across which there is drawn at each layer of yarn a thin screen of parallel fibers to produce a series of 30 or more interlaced filter sheets, each securely locked in place by the subsequent layer of yarn.

The liquid to be filtered is forced through these tunnels from outside to inside of the tube, and accordingly passed through a manifold series of filter sheets. Rugged construction and special sealing of the ends of the tube when placed in the filter prevents bypassing to insure positive filtration, according to the maker.

The tubes are made with various densities of fiber in the filtering tunnels for liquids of various viscosities and for varying degrees of cleaning desired. The filters may be used in parallel for filtering large volumes or in series, ranging from coarse to fine. The multi-tube models are available on a stationary base, or on a movable truck with pump and motor attached.—The Cuno Engrg. Corp., Meriden, Conn.

Pressure, Relief Control for Hot Water Heating

No. 306—The combination pressure and relief control illustrated is used for keeping a hot water heating system automatically filled with water, and for automatically relieving any

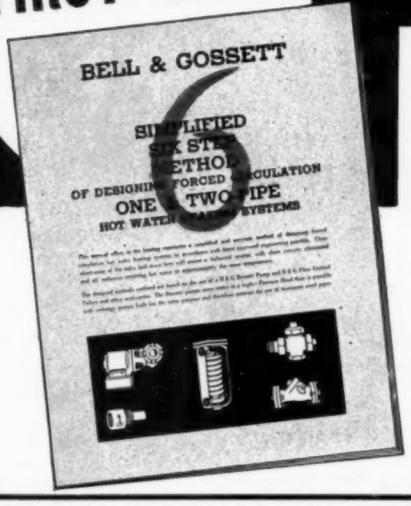


excess pressure that may occur. Capacity of one model is extra large, and the manufacturer recommends it when quick filling of the system is desirable.

It consists of a relief valve in combination with a reducing valve, made of steam bronze. Both valves have large laminated phosphorus bronze diaphragms sensitive and positive in action which resist heat and corrosion to retain original flexibility. Gaskets above and below diaphragms and relieved casting corners

for the FIRST TIME

A SIMPLIFIED DESIGN MANUAL For HOT WATER HEATING SYSTEMS!



HEATING CONTRACTORS! HERE IS THE BOOK WHICH ENABLES YOU TO SELL RADIATOR HEAT AGAINST TODAY'S COMPETITIVE CONDITIONS.

Nothing like it ever published! Bell and Gossett leads the way again with *the first authoritative instruction book on designing and pipe-sizing forced circulation Hot Water Heating Systems.* The latest approved engineering practice reduced to six simple steps—written so anybody can understand and use it. 20 pages of sketches, handy tables, and other data exceedingly valuable to the heating contractor.

Now . . . the smallest pipe and radiation sizes ever possible

This book shows you how to design forced circulation hot water heating systems with piping and radiation of practically steam sizes. Instructions cover both Direct and Reverse Return two-pipe systems and also full information on the B. & G. Mono-Flo Fitting—a device which completely revolutionizes one-pipe hot water systems.

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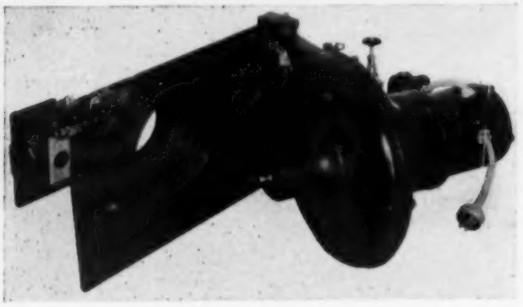
are designed to prevent binding and assure diaphragm efficiency.

Discs are special composition of proper hardness supported in disc holders by large head screws. The only spring exposed to water is the one below the seat in the reducing valve. Brass strainers protect seats of both valves, and relief valve has lever for testing in service without building up a pressure in excess of valve setting.

The standard model is similar except for capacity. Pipe size is ½ in.—Penberthy Injector Co., Detroit, Mich.

Horizontal Rotary Type Oil Burner

No. 307—New oil burner shown is horizontal rotary type, using No. 5 oil under automatic control and No. 6 oil employing



preheating. It is made in four sizes, varying from 15 to 70 gallons per hour, or from 6000 to 28,000 sq ft of steam radiation. —Ace Engineering Co., 208 W. Washington St., Chicago, Ill.

Add the new products and new companies listed in "Equipment Developments" to your Directory Section (which you received in your January 1935 HEATING, PIPING and AIR CONDITIONING) and thus keep your records of sources of supply up-to-date throughout the year.

Conventions and Expositions

American Oil Burner Association: Annual convention, April 8-9, Claridge Hotel, Atlantic City, N. J. Headquarters Office, 342 Madison Ave., New York, N. Y.

Heating, Piping and Air Conditioning Contractors National Association: Annual convention, May 20-22, Hotel Gibson, Cincinnati, Ohio. Acting Secretary, S. Lewis Land, 1250 Sixth Ave., New York City.

American Society of Refrigerating Engineers: Spring convention, May 22-24, Statler Hotel, Detroit, Mich. Secretary, David L. Fiske, 37 W. 39th St., New York City.

Coal Utilization Short Course: June 11-13, College of Engineering, University of Illinois, Urbana, Ill. Under auspices of the departments of mining and metallurgical, and mechanical engineering.

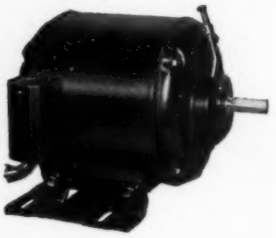
National District Heating Association: Annual convention, June 11-14, Bellevue-Stratford Hotel, Philadelphia, Pa.

American Society of Mechanical Engineers: Semi-annual meeting, June 19-21, Cincinnati, Ohio. Secretary, C. E. Davies, 29 W. 39th St., New York City.

American Society for Testing Materials: Annual meeting, June 24-28, Book-Cadillac Hotel, Detroit, Mich. Exhibit of testing and related equipment to be held concurrently. Assistant Secretary, R. E. Hess, 260 S. Broad St., Philadelphia, Pa.

American Gas Association: Annual convention, October 14-18, Atlantic City, N. J.

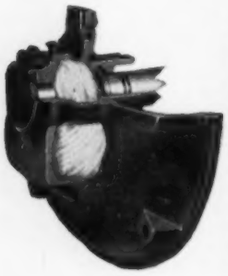
Wagner Motors are Dependable



THERE was a time, you will probably recall, that if you drove your automobile a distance of fifty miles without tire or machine trouble you considered yourself lucky. Auto users in those days knew that their cars were not very *dependable* and were prepared for emergencies. Those days are over! It's *dependability* auto owners desire today.

Today the public is demanding air-conditioning equipment that is *dependable* and will operate efficiently for many years.

Wagner engineers with forty-four years of experience in building and servicing motors have developed a complete line of *dependable* air-conditioner motors.



Among the distinctive features contributing to the *dependability* of Wagner motors is the steel-backed babbitt-lined bearings with wool-yarn lubrication — steel-backed to withstand all strains to which bearings are subjected — babbitt-lined to prevent "bearing seizure." The wool-yarn is used to carry on an uninterrupted supply of filtered oil to all parts of the bearing. The upper photo shows part of bearing housing and oilwell. A portion has been cut away to reveal construction details.

Another outstanding feature is the Wagner stator — strong and rigid, yet mechanically simple. The lower photo shows a view of a stator core and frame. The frame is of rolled steel—strong, rigid, will not get out of alignment—unbreakable. Stator coils are completely insulated from frame — securely wedged in place by special tough fibrous slot insulation. Complete stator core is baked in drying ovens to drive out all moisture, then while still hot it is immersed in heavy insulating varnish, selected for its penetrating and insulating qualities, to thoroly impregnate the windings.

Dependability is but one of several qualities that air-conditioner manufacturers must consider. Other considerations fully provided for in Wagner design and construction are: quietness, interchangeability, appearance, and performance. For a complete description of Wagner motors ask for Bulletin 167.

There's a Wagner motor to fit your needs, regardless of the type you prefer. Included in the Wagner line of motors are

Type RB	Split Phase
Type KA	Repulsion-Start-Induction
Type RBZ	Capacitor-Start Induction-Run
Type RZH	Capacitor-Start Capacitor-Run
Type RD	Direct-Current
Type RP	Polyphase Squirrel-Cage

Wagner Motors are available in such various mechanical variations as open, totally-enclosed, or drip-proof; rigid or resilient mounted; horizontal, vertical, or flange-mounted; sleeve or ball bearing, etc.

MOTORS
TRANS-FORMERS

Wagner Electric Corporation
6400 Plymouth Avenue, Saint Louis, U.S.A.

FANS
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HEATING OR COOLING SURFACE

● There is nothing more modern than Aerofin Standardized Light-Weight Fan System Heat Exchange Surface.

For either Cooling or Heating applications, Aerofin is the choice of important architects, engineers and contractors because it gives complete satisfaction and unvarying performance.

Aerofin begins where ordinary surface leaves off. Its exclusive features in design and construction make it the most advanced surface available today. It is furnished in aluminum, copper or other special metals.

The home office in Newark or any branch office will gladly send descriptive literature or render prompt, personal and efficient technical cooperation. Simply write to the address below.

Aerofin Flexitube Unit for Heating or Cooling with water.

The Shrine of The Little Flower, Royal Oak, Mich. has Aerofin heating surface. Architect, Henry C. McGill, New York; Consulting Engineer, William Brown, Detroit; Heating Contractor, Fred J. Douglas, Birmingham, Michigan.

AEROFIN is sold only by Manufacturers of Nationally Advertised Fan System Apparatus.

Literature upon Request

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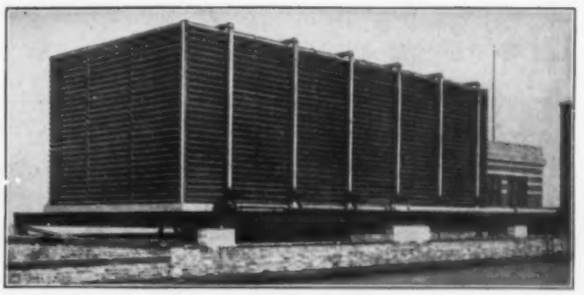

Over 3000 BINKS Cooling Systems Now in use!

BINKS ATMOSPHERIC SPRAY COOLING TOWERS are built in sizes ranging from 6 tons up to 200 tons refrigeration capacity, and are extensively used today in the ever growing air conditioning field.

BINKS INDOOR FORCED DRAFT Cooling Towers are also made up to 40 tons refrigeration capacity and are unexcelled for the smaller tonnage requirements where piping difficulties are encountered and piping cost must be minimized.

We offer a standardized complete SHOP CONSTRUCTED TOWER (with full spray nozzle distribution) for most every application. This is evidenced by the more than 3,000 Binks Cooling Towers now in operation.

The Binks line is recognized as being the most complete and includes spray and atomizing nozzles of advanced design for practically every industrial requirement.

A FEW OF THE 3000

Goodyear Tire & Rubber Co.
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Firestone Tire & Rubber Co.
Brown Shoe Company
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Brewery Corp. of America
Roosevelt Hotel Co.
University of Michigan
Red Top Malt Extract Co.
Buda Company
Georgia Power Co.
East Ohio Gas Co.
Sheffield Farms Co.
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Great Lakes Forge Co.
Carrier Engineering Corp.
First National Stores
United States Gypsum Co.
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Send for Bulletin 60 On Atmospheric Towers 5 to 1200 G.P.M.

Send for Bulletin 70 on Indoor Forced Draft Towers 5 to 150 G.P.M.

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Book Reviews

Three More Air Conditioning Lessons

"Air Conditioning for Heating Contractors" (Part 5), published by Heating, Piping and Air Conditioning Contractors National Association. 1250 Sixth Ave., New York, N. Y. 29 pp., 4¼x7¼ in., paperbound. Price, 50c to non-members.

The three lessons included in Part 5 complete the series of 18 prepared and published by the committee on air conditioning of the *H.P.A.C.C.N.A.* and edited by S. Lewis Land. The three concluding lessons are Theory of and Equipment for Steam Jet Refrigeration; Selection and Design of Steam Jet Refrigeration Equipment, both by L. L. Simmons; and Application of and Equipment for Ice Cooling, by John Everetts, Jr.

Each of the lessons is a clear, concise discussion of the respective subject.

Air Conditioning and Engineering Manual

"Air Conditioning and Engineering," by the engineering staff of American Blower Corp. Published by American Blower Corp., 6000 Russell St., Detroit, Mich. 704 pp., 8½x11¼ in., clothbound. Price, $5.00.

Compiled principally for use of the publisher's own organization and general reference purposes, this 704 page book is a comprehensive exposition covering the entire field of air conditioning and engineering. Numerous charts, photographs, tables, and drawings are included; 332 pages comprise an apparatus and equipment section with dimension and capacity tables and other data on air conditioning and air handling equipment.

The technical section is divided into 23 chapters on the following subjects; Air, heat and steam, air flow, air ducts, fans, power selection for fan drive, sound, ventilation, heating, unit heaters and ventilators, humidifying, cooling and dehumidifying, refrigeration, temperature and humidity control, vapor absorption and removal, drying, ceramics, furnaces, mechanical draft, fume collection and removal, dust collection and pneumatic conveying, mine ventilation, and tables and conversion factors.

According to the publisher's announcement, this is really the second and corrected printing of this book, as 100 copies were placed in the hands of the authors and men in the field six months ago for checking. Over four years were required to write and compile the volume, it is said.

Air Conditioning Service Manual

"Official Air Conditioning Service Manual," by L. K. Wright, published by Gernsback Publications, Inc., 99 Hudson St., New York, N. Y. 347 pp., 8½x11 in., in looseleaf post binder.

The author, senior refrigeration instructor at the West Side Y. M. C. A. Trade and Technical School of New York, presents in this manual material on the fundamentals, estimation, installation and servicing of the smaller unit air conditioning apparatus. Most of the 35 chapters into which the manual is divided are devoted to refrigeration for air conditioning.

Following a brief review of the history of air conditioning, theory and fundamental laws are considered, followed by a discussion of methods of refrigeration, the ejector system, and the compression system. A chapter each is given to refrigerants, lubricating oils, and liquid throttle devices. A chapter on the field of air conditioning follows, as do sections on thermodynamics, weather, brines, control devices, and the handling, testing and storage of refrigerants.

Servicing is covered in chapters on refrigerating systems, expansion and float valves, tools for servicing and installation, and

Harborview Hospital, Seattle, Washington
Architects, Thomas, Grainger and Thomas

3 centrally located tank rooms are used to supply Nitrous Oxide, Oxygen and Ethylene to 23 operating rooms throughout the hospital.

"SLEEPING GASES" for Patients Only

The Hays Copper Plumbing Method was selected for piping "Sleeping Gases" to 23 Operating Rooms in this $2,750,000 Hospital. The elimination of unnecessary fittings and the mechanical superiority of the compression type joint were important reasons for the selection of the Hays Copper Method.

Mr. D. P. Thomas recently reported:

"It was particularly necessary to have joints that would insure against leakage as any losses in the gas lines would produce a fire hazard and expensive gas losses. A test pressure of 100 lbs. was maintained for 48 hours on the system prior to acceptance and no leaks whatever were recorded. This system has been in operation for about 4 years and has given no trouble whatever."

Double Seals seize copper pipe in a doubly tight grip. A 45° plus a 90° flange on the pipe and a 45° plus a 90° seat on the fitting together give great mechanical strength to this joint. Send for catalog and test data.

Available in 400 Styles and Sizes

Approved by Underwriters Laboratories

Specify Hays Double Seals for Sturdy and Trouble-Free Copper Connections.

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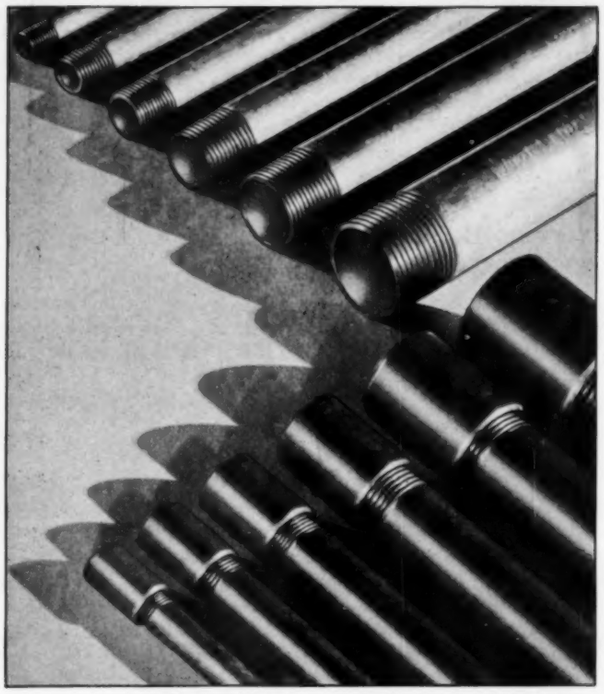

..A BETTER STEEL PIPE

The Fretz-Moon "continuous process" differs radically from other methods of pipe manufacture. And, likewise, Fretz-Moon Pipe differs from ordinary steel pipe, because it is better.

Made of highest quality steel skelp that is heated throughout every foot at unvarying temperature, for a definite period of time, without contamination from furnace bottom, and then roll-welded, sized, cut and cooled under precision control, this modern steel pipe possesses high degrees of uniformity, ductility and cleanliness not found in steel pipe made by ordinary methods.

If you would like to read the detailed story of the Fretz-Moon "continuous process," write for it.

FRETZ-MOON TUBE CO., INC.
BUTLER • PENNA.

FRETZ-MOON PIPE
BLACK AND GALVANIZED

Cornell University Regulates its Cooling System with Sarco Temperature Regulators

To control the flow of brine into the coil of each of the fifteen air-conditioned rooms in the attractive new million dollar Cornell Plant Science Building, at Ithaca, N. Y., Sarco Temperature Regulators are used. This assures the desired constant temperature.

You can set the Sarco for any desired temperature from zero up and it will maintain exactly that temperature automatically without any attenion.

SARCO TEMPERATURE REGULATORS

The reason for its accuracy is that the Sarco is actuated by liquid expansion, which gives more power to operate the valve and assures the same sensitivity at all points of the range.

Sarco Temperature Regulators are inexpensive to install and maintain because no compressed air, electricity or water pressure is needed for their operation.

We'll send one on 30 days' free trial to test in your cold storage rooms, pasteurizers, drying rooms, kilns, hot water heaters or elsewhere.

Write for Catalog O-52

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183 Madison Ave. New York, N. Y.

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IRON *and* **BRASS**

First of all, you insist on getting your dollar's worth of fittings and nipples.

But, as a careful buyer, you look beyond the number of pieces of iron and brass in your bins. You want your merchandise to contribute to the honesty and good will of your business. You buy more than merchandise.

You buy dependability. It is a quality which keeps your old friends and makes new ones. In fittings and nipples, it means installations made in record time without waste of material. It means installations which don't spring leaks after the pressure is turned on. It means one less thing to gamble on and worry about.

Dependability is built into Devlin Fittings and Nipples as surely as is the size and style and material. It has caused our business to grow steadily on a firm basis — and it will do that for yours, also.

For dependability, be sure to

specify **DEVLIN**

FITTINGS IN MALLEABLE IRON, CAST IRON, AND BRASS
NIPPLES IN STEEL, WROUGHT IRON, BRASS AND COPPER

THOMAS DEVLIN MANUFACTURING CO.
Incorporated
BURLINGTON, N. J.

motor troubles, tests and repairs. Refrigerant hazards, insulating materials, and heat transmission through walls each are given a section.

One chapter each is devoted to estimating a small store, a restaurant, a home, and layout of a duct system. Following sections on starting up a system and operating and servicing, the manual concludes with five chapters describing various types of air conditioning equipment, and a miscellaneous data section.

Numerous tables and illustrations drawn from a variety of sources are included.

German Book on Public Building Ventilation Reviewed

"The Ventilation of Public Buildings" (Die Lüftung von Aufenthalts- und Versammlungsräumen), edited by H. Behrens. Published by "Heizung und Luftung," Berlin SW 29, Bergmannstr. 56. 186+ x pp., 8½ x 11 in.; cloth bound, 5½ RM; paperbound, 4 RM.

Reviewed by F. E. Giesecke.

This book is divided into twelve sections which were prepared by leading specialists in heating and ventilation or related sciences. It represents the papers and discussions presented at a meeting of the association of German heating engineers.

In Section I—Introduction—Dr. Marx discusses the development of the art of ventilation in Germany and divides it into three periods. During the first period artificial ventilation was installed extensively; during the second period the medical fraternity favored natural ventilation through open windows; and during the third and present period, artificial ventilation is again given preference.

Dr. Gruenewald discusses the hygiene of heating and ventilation and recommends definite values.

In Section II—The Cleaning of Air—Dr. Wittemeier, Director of Delbag, points out that pure air is fully as important to human well-being as pure water and that while pure water is now generally supplied, much progress must still be made to secure pure air. He discusses the various types of dust particles and their effects on human beings. He shows photographs of the Merchandise Mart of Chicago, Empire State Building, the Waldorf-Astoria, and the Columbia Presbyterian Medical Center of New York, all equipped with air filters, and deplores the fact that the installation of air filters is proceeding much more slowly in Germany than in the United States.

In Section III—Ventilation of Hospitals—Director Taubert discusses the subject from the viewpoint of the ventilating engineer, and Dr. Petrick from the viewpoint of the architect. Mr. Taubert states that hospitals are now being built in Germany without means for artificial ventilation and that German hospitals which were built about 30 years ago and are equipped with means for artificial ventilation are being operated without the use of their ventilating systems. He calls attention to the importance of providing patients in hospitals with pure air of proper temperature and humidity and states that the cost of installing and operating modern ventilating systems is not prohibitive.

Dr. Petrick estimates the cost of a ventilating system at from 1½ to 2 per cent of the total cost of the building and discusses the cost of maintenance and operation and concludes that these costs would be permissible if a really satisfactory installation could be secured. He concludes that the ultimate development will probably be a combination of central ventilation and window ventilation and recommends a patented type of window which is described in detail.

The discussions of these papers evidently brought out widely diversified views. For example, Dr. Dosquet suggests (page 43) that heating engineers should confine their activities to the ventilation of cafes and restaurants and leave the ventilation of hospitals to the medical profession.

In Section IV—The Ventilation of Schools—Commissioner Nydahl discusses this subject from the viewpoint of the teacher,

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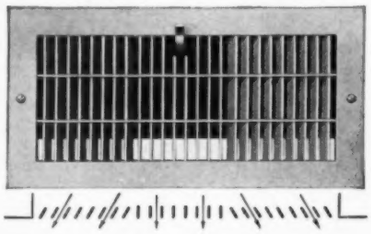

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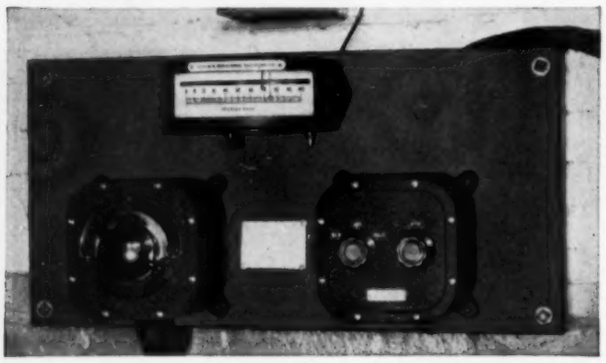

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Dr. Drigalski from the viewpoint of a sanitarian, Professor Winterstein from the viewpoint of an architect, and Commissioner Schmidt from the viewpoint of the heating and ventilating engineer.

Commissioner Nydahl stresses the importance of good ventilation, not only to protect the health of the children, but also to increase their ability to learn. He does not express preference for either artificial or natural ventilation, but suggests that all new schools be built in suburbs or parks where the air is comparatively pure and that all new school buildings be built of one-room depth so that good ventilation can be secured by open windows and doors. He calls attention to the open-air schools in Berlin and suggests that if this type of building is good for sickly children it must be even better for healthy children.

Dr. Drigalski discusses the technical details of ventilation and states that from 5 to 10 per cent of all children are carriers of infectious germs to which they themselves are immune but which attack others. He stresses the importance of admitting as much sunlight as possible into the rooms since short wave length radiation weakens or kills even very resistant micro-organisms quickly. He states that it has been shown that a considerable portion of the bactericidal portion of the sun's rays penetrate city air and the double windows of Berlin school houses even during autumn days.

Professor Winterstein gives detailed information and recommends artificial ventilation as a supplement to natural ventilation.

Commissioner Schmidt recommends natural ventilation supplemented where possible by artificial ventilation and explains both systems in detail.

The discussion was concluded by Dr. Groeber with a recommendation that the engineers make a careful study of the ventilation of school buildings and after they have agreed among themselves on the best plans, that they request permission to execute the plans in some of the future school buildings in order to have an exemplary installation with respect to efficiency, simplicity, and economy. (Dr. Groeber is Professor of Heating and Ventilating at the Technical High School at Charlottenburg and is the author of the latest edition of Professor Rietschel's text on heating and ventilation).

In Section V—The Ventilation of Public Assembly Rooms—is discussed by Chief Engineer Metzkow, who describes methods of determining the quality of air, and discusses the relative merits of downward and upward ventilation and recommends upward ventilation because it is more economical in operation costs, more sanitary, and, in case of fire, very much safer. He cites as an objection to downward ventilation that the exhaled air has a temperature of about 99 degrees, and, by virtue of this higher temperature, rises and mixes with the cooler incoming air and then returns to be re-inhaled in a diluted form. He presents descriptions and illustrations of the ventilating systems of theaters and large exposition halls.

In Section VI—The Ventilation of Hotels and Similar Buildings—is discussed by Dr. Marx, who calls attention to the scientific principles involved and urges the installation of efficient ventilating systems in all cases.

In Section VII—Power Ventilators—Chief Engineer Fuchs discusses centrifugal ventilators with vanes of various shapes and shows performance curves of several types. He recommends the use of fans with blades curved backward because they are more efficient, the power required is practically independent of the volume of air moved, and the pressure decreases as the volume of air increases, which facilitates the operation of several fans in parallel.

Commissioner Behrens discusses propeller type fans and suggests that the ventilating engineer should design his equipment to permit the use of propeller type fans wherever possible because of their greater efficiency.

Director Ziehl discusses electric motors with particular reference to noiseless operation and describes how he began, about

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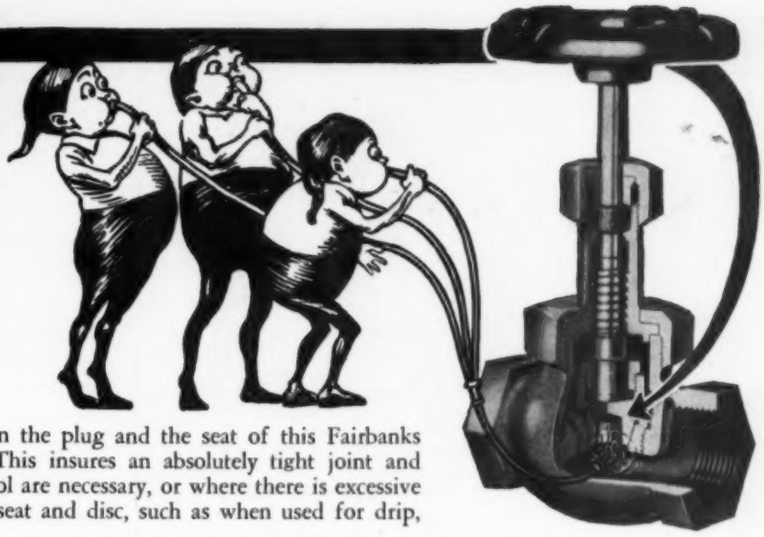

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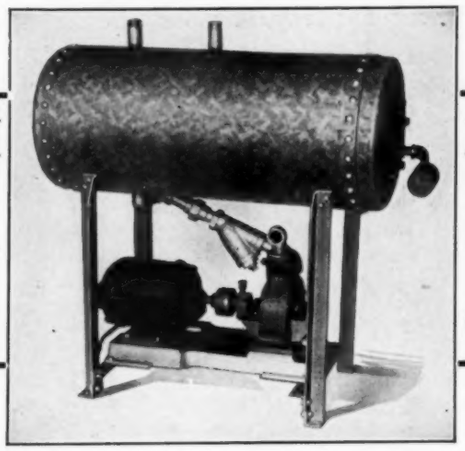

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ten years ago, to build motors for use with church organs which are absolutely noiseless. The principal difficulty arose in the elimination of magnetic noises; this was accomplished by building the motors so that the magnetic oscillations exceeded 4,000 per second, the limit audible to the human ear.

In Section VIII—Construction and Design of Air Heaters and Coolers—Chief Engineer Thomas describes and illustrates various heat exchangers from steam, water, gas, and electricity to air; discusses their relative economy in operation and states that if gravity flow water is used instead of steam, the heat exchange is about 20 per cent lower.

In Section IX—The Design of Ventilating Pipes and Ducts is discussed by Commissioner Behrens in a manner similar to that presented in the Rietschel-Groeber textbook on Heating and Ventilation.

In Section X—Appliances for the Measurement and Regulation of Air Conditioning Apparatus—discussed by Chief Engineer Mattner and Dr. Liesegang:—The former traces the historical development of instruments for measuring, recording, and regulating apparatus used in air conditioning and describes many modern appliances. Dr. Liesegang describes electrical apparatus used for indicating, recording, and regulating air conditioning appliances. The instruments are fully illustrated and described in detail and are evidently very efficient in their operation.

In Section XI—Automatic Appliances for Air Conditioning—are discussed and illustrated by Engineer Gieg and their uses illustrated by numerous examples.

In Section XII—Conclusion—Commissioner Behrens summarizes the information which an architect should have regarding the ventilation of buildings.

As a whole, the book presents the views of leading German experts and is well worth the careful study of engineers and others interested in the ventilation of buildings.

Booklets, Reports and Papers

Silica Gel for Industrial Processing Applications

"Silica Gel and Its Uses in Conditioning Air and Gases for Industrial Processing," by J. Maynard Lednum. Published by industrial gas section of American Gas Association, 420 Lexington Ave., New York, N. Y. 14 pp., 6¾x10 in., paper cover.

Object of this paper is to present typical data on adsorption of gases and vapors by silica gel and to show various arrangements of equipment which are applicable to the many processing operations where either low or controlled humidities are required. The author points out that practically all process operations in manufacture of materials require the removal of a liquid from solids, and that such removal occurs in three stages; (1) initial adjustment period; (2) constant rate period; and (3) falling rate period. In the third stage, a surrounding atmosphere of low humidities or low vapor concentrations is often required; for instance processing of hygroscopic materials such as paper, artificial silk, cellophane, leather, cereals, chemicals, pharmaceuticals, dyes, etc., require third stage conditioning.

Maintenance of the proper low humidity atmosphere may be obtained by refrigeration, chemical action, drying in *vacuo*, or adsorption. Action of an adsorbent is described, and the silica gel system and silica gel itself discussed. Theory of adsorption is discussed in detail, activation of the gel described, and action of the gel explained.

Applications of the silica gel system are given specific attention, nine different arrangements being diagrammed and explained. The author points out in conclusion that in addition to high adsorptive powers, it is necessary to take into consideration

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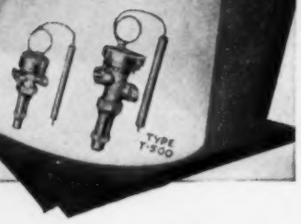

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A well selected bibliography is appended.

Water as an Engineering Material

"Water as an Engineering and Industrial Material," by Sheppard T. Powell. Published by American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa. 45 pp., 6x9 in., paper cover. Price, 50c.

This is a reprint of the Edgar Marburg lecture delivered by Sheppard T. Powell, consulting chemical engineer, at the 1934 annual meeting of the *A.S.T.M.* at Atlantic City, N. J. It is of particular value to those concerned with industrial applications, boiler plants, water analysis and detection of impurities, but it is also pertinent to the general water supply field.

The lecturer presents current thought on such problems as removal of suspended solids from water, effect of color on industrial water supplies and detrimental effect and removal of manganese. A considerable portion of the lecture discusses the treatment of water for use in boilers. The application of petrographic examination of boiler scales is indicated and the characteristic properties of minerals are described.

The section on influence of concentrated boiler-water salines on cracking of boiler steel and prevention of caustic embrittlement is important in view of the failure of many boilers from this cause.

Also discussed are the corrosion of iron and steel in steam generating systems and the reduction of corrosion by deaëration of water. The discovery by spectrographic analysis of lead, fluorides, zinc, and other metals in appreciable amounts in municipal water supplies is evaluated.

Effect of Soot on Boiler Heat Transmission

"Effect of Soot on Heat Transmission in Small Boilers," by P. Nicholls and C. E. Augustine, Bureau of Mines Experiment Station, Pittsburgh, Pa. Mimeographed pages, 8½x11 in.

The United States Bureau of Mines has issued report No. 3272, "Effect of Soot on Heat Transmission in Small Boilers," by P. Nicholls, supervising engineer, fuels section, Pittsburgh Experiment Station, and C. E. Augustine, associate fuel engineer of the same station. This investigation was undertaken because the Bureau of Mines frequently receives requests for authoritative information on the effect of soot in small and other heating boilers. A number of tests were conducted which shows that the insulating effect of deposits of soot decreased the heat absorbed by the boilers 2 to 7 per cent of the heat made available by the fuel, depending on the thickness of the deposit. However, the losses, expressed as a percentage of the coal burned, will be less and will be approximately proportional to the over-all efficiency. Thus, with 60 per cent efficiency, about 1¼ to 4¼ per cent of the coal will be wasted because of the insulating effect of the soot.

The Committee of Ten—Coal and Heating Industries, 307 N. Michigan Ave., Chicago, Ill., has reprinted this report as Miscellaneous Report No. 12, "Effect of Soot on Heat Transmission in Small Boilers." Copies are available to all contributing members without charge; to others for 5c.

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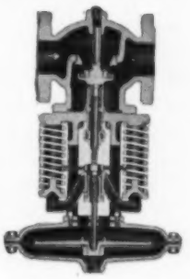

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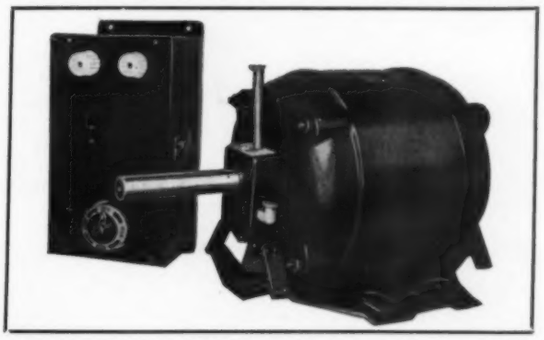

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Also, a bulletin giving complete data—sizes, shapes, and capacities—for filters for compressors, internal combustion engines, etc.

No. 402. CIRCUIT BREAKERS: General Electric Co., Schenectady, N. Y. Four page bulletin on trip free air circuit breakers for buildings, plants and power stations, manually or electrically operated, 600 volts, 1600-6000 amp a-c; 750 volts, 1600-10,000 amp d-c. Features, dimensions, and ratings are given.

No. 403. COILS: Fedders Mfg. Co., 57 Tonawanda St., Buffalo, N. Y. 28 page technical booklet on standard cooling and dehumidifying coils for air conditioning. Seventy-five rating tables for as many different coils are presented, basis of ratings is explained, typical details and connections are shown, and graphs for vapor pressure of various refrigerants, heat content of air and moisture, and logarithmic mean temperature difference are included, as well as a psychrometric chart.

No. 404. DRIVES: Link-Belt Co., 910 S. Michigan Blvd., Chicago, Ill. 32 page pictorial booklet illustrating numerous examples of applications of "Silverstreak" silent chain drives. Features of the drives are carefully explained.

No. 405. FANS: B. F. Sturtevant Co., Hyde Park, Boston, Mass. 66 page catalog of "Silentvane" fans (Design 5) with backward curve blades for ventilating systems in large public buildings, auditoriums, vehicular tunnels, and many industrial applications. Following a brief general description and views of typical applications, 40 pages of performance tables for fans from 15¼ to 97½ in. wheel diameter, single and double width, are given, standard designation of fans is explained, and standard fan arrangements are shown. Dimensions and specifications are included.

Also, a similar 44 page catalog for "Multivane" fans (Design 6) with forward curved blades, recommended particularly by the maker for heating, ventilating and air conditioning of buildings such as schools, hotels, office buildings, etc. Sizes from 12 to 87 in. wheel diameter, both single and double width, are covered in this catalog.

No. 406. GEARS AND SPROCKETS: The Medart Co., 3500 DeKalb St., St. Louis, Mo. 80 page catalog of gears and sprockets and power transmission equipment comprising tables of specifications, list prices, and other information.

No. 407. HOT WATER HEATING: Bell & Gossett Co., 3000 Wallace St., Chicago, Ill. 20 page design manual available to heating contractors, architects, and engineers for forced circulation hot water heating systems. The instructions given base design on six steps, feature being the use of piping and radiation of practically steam sizes, possible through use of this company's centrifugally designed circulating pump. Heat loss coefficients and instructions for computing radiation are included.

No. 408. METAL BENDINGS: Wallace Supplies Mfg. Co., 1310 Diversey Pkwy., Chicago, Ill. Eight page bulletin illustrating metal bendings of steel, brass, copper and aluminum—pipe, tubes and other sections—including quarter bends, return bends, offset bends, and numerous others for refrigeration and steam coils and a variety of other applications.

No. 409. PLATES: Lukens Steel Co., Coatesville, Pa. Eight page booklet presenting a pictorial summary of the world's largest plate mill and its products, including nickel clad steel and other products.

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Also, six page bulletin of centrifugal pumps, Types C, Ca and CB.

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No. 413. Independent Register & Mfg. Co., 3757 E. 93rd St., Cleveland, Ohio. 24 page catalog of registers, ventilators, and grilles, with list prices.

No. 414. VALVES: Parker Appliance Co., 10320 Berea Rd., Cleveland, Ohio. 60 page booklet covering typical variety of valves with tube coupling machined outlets, 2 in. O. D. and under, made by this company. Drawings, complete information, and list prices are given for the many styles shown.

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